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NA-65-825-1

ENERGY STORAGE SUBSTATION CONCEPTS FOR AIRCRAFT ACTUATION FUNCTIONS

SECOND QUARTERLY TECHNICAL REPORT

(THIS REPORT IS PREPARED IN COMPLIANCE WITH CONTRACT AF33(615)2971 FOR THE AIR FORCE AERO PROPULSION LABORATORY)

NORTH AMERICAN AVIATION, INC. / LOS ANGELES DIVISION

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Serial No. 25

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ENERGY STORAGE SUBSTATION CONCEPTS

FOR

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Contract AF33(615)-2971 for the Air Force
Aero Propulsion Laboratory)

BPSN No. 5 (63-8128-624, 5214)

APPROVED BY

R. [illegible] Dawson
Program Manager

20 January 1966

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FOREWORD

This is the second quarterly report concerning research and development of the flywheel as an energy storage substation concept for aircraft actuation systems. The period of effort which this report documents is from October 31, 1965, through December 31, 1965.

This program under Contract AF33(615)-2971 is being conducted by North American Aviation Inc., at the Los Angeles Division.

Publication of this report does not consistute Air Force approval of the report's findings or conclusions. It is published only for the exchange and stimulation of ideas.

Air Force Aero Propulsion Laboratory, RTD
Wright-Patterson Air Force Base, Ohio

This program is being monitored by Lt. R. N. Alexander of the Aero-Propulsion Laboratory. The program is being conducted at North American Aviation Incorporated, Los Angeles Division with Mr. R. J. Dawson as Program Manager, Mr. C. W. Helsley as Principal Investigator and with the assistance of Mr. C. Simpson, Mr. B. Call, and Mr. C. Crother, on analysis evaluation, and testing.

ABSTRACT

The first quarterly report for this program established design criteria and performance data for evaluation of the flywheel as a component. The portion of effort covered by this second quarterly report includes the studies designed to integrate the flywheel into existant aircraft actuation systems and evaluate the weight, reliability, and performance characteristics which result.

One of the major factors which reduces the accuracy of this evaluation is the fact that hardware is not developed and tested which allows accurate definition of the flywheel housing, evacuation system, mounting and gearing in terms of weight, space, and compatibility with the air vehicle environment and actuation system components.

Further in the course of the system investigations, several apparent facts were brought to light which tended to suggest an unanticipated necessity to direct development effort, in the exploitation of flywheel energy storage, toward utilization of all-mechanical couplings between the flywheel input and output components. The development level of these couplings whether they be, direct, on-off clutches, or servo controls, will establish an important measure of the probable successful application of the energy storage substation, in reducing air vehicle weight, within the minimum reliability limits that are acceptable.

Performance of these studies revealed many basic relationships which are essential to the evaluation of the use of the flywheel as an energy storage substation that would not otherwise have been apparent.

The studies covered two basic areas which were, continuous duty cycle operation as typified by the XB-70 elevon system, and intermittent duty cycle operation as typified by the XB-70 landing gear system. The elevon system studies compared a hydraulic powered energy storage substation and an energy storage substation with hydraulic power input and mechanical power extraction to the system currently in use on the XB-70. In each instance the compared systems equaled the basic system in reliability and were judged in some areas to exceed the basic system in performance.

The measure of comparison then became weight. On this basis the comparative systems turned out as follows:

1.	BASIC SYSTEM	2005.5 lbs
2.	ALL HYDRAULIC ENERGY STORAGE SUBSTATION SYSTEM	1848.7 lbs
	WEIGHT SAVINGS	156.8 lbs
3.	HYDRAULIC INPUT MECHANICAL OUTPUT ENERGY STORAGE SUBSTATION SYSTEM	1539.0 lbs
	WEIGHT SAVINGS	466.5 lbs

v

The landing gear system studies used essentially the same approach. In this case the comparative weights became:

1.	BASIC SYSTEM (Weight of items which change weight)	612 lbs
2.	ALL HYDRAULIC ENERGY STORAGE SUBSTATION SYSTEM	600 lbs
	WEIGHT SAVINGS	12 lbs
3.	BASIC SYSTEM (Weight of items which change weight)	1503.1 lbs
4.	HYDRAULIC INPUT MECHANICAL OUTPUT ENERGY STORAGE SUBSTATIONS	1190.8 lbs
	WEIGHT SAVINGS	312.3 lbs

Analog computer data accumulation runs were completed on the F-100 horizontal stabilizer system. However, the data have not been completely reduced and digested so the final conclusions to be drawn from this effort will be included in the final report.

SYSTEM ANALYSIS

In conformance with the terms of the contract numerous systems incorporating the flywheel as the energy storage device have been studied. Also in conformance with the contract the bulk of the effort has been concentrated on configurations 1 through 4, page 22 of NA-65-124. Each of the systems corresponding to these configurations is basically hydraulic but involves some variations of component arrangements. The one exception to this general rule is the fact that configurations 3 and 4 call for the use of a mechanical rotary hinge as the power output device in place of the conventional hydraulic linear actuator.

It was also planned to direct at least half the studies towards a relatively small airplane such as the F-100 so that comparisons could be made between it and larger aircraft. However, it shortly became apparent that, when the relatively low power requirements of this air vehicle were coupled with the need for multiple substations to meet redundancy requirements, the individual substations became very small. Flywheels on the order of a 3-inch diameter turning at 200,000 rpm powered by hydraulic motors smaller than any existing frame sizes seemed to be what was required. In view of the problems involved in arriving at a fair weight comparison when dealing with subminiature components operating at speeds outside conventional component experience it was decided to concentrate on the XB-70 where systems of more reasonable size would be required. As a result the studies were concentrated on powering the XB-70 elevon system and the XB-70 landing gear system.

The problem of working with components of practical size, however, did not have an effect on the analogue computer studies. These studies continued using the F-100 horizontal stabilizer at 3000 PSI system pressure. The studies were based on flywheel power input and output shafts that have maximum angular velocity of 100 radians/second. The conversion to higher angular velocities by assumed gearing, has the effect of reducing the moment of inertia and weight at the expense of greater stresses in the flywheel material. In summary the study efforts which were oriented toward practical hardware considered XB-70 systems, while those which were oriented towards control system characteristics continued considering F-100 requirements as was originally planned in the contract.

The accomplishement of these studies revealed several basic relationships between substation components and pointed to possible considerations that would not have been apparent had it not been for the studies. Several of these significant facts are as follows:

1. Transmittal of losses from source of power to remote station - Conventional arrangements for driving hydraulic pumps from the engine pad, discard the pump inefficiency losses as pump case and fluid temperature increases. If the flywheel energy storage concept should relocate a portion of the pump capacity to a remote position, these losses would be transmitted to the remote position before being discarded as heat.

The transmission of these losses requires increased trunk line capacity and reduces the benefits derived from the flywheel substation concept. This same principle is applicable for cases where the transmission from the engine to the remote station is mechanical, electrical, or pneumatic.

2. Pump capacity versus maximum system demand - By application of the flywheel substation concept to conventional hydraulic systems which have engine driven pumps sized equal to or larger than the maximum system flow demand, a reduction in pump size at the engine can be effected. The reduction, however, does not reduce the total pumping capacity, therefore, does not reduce total pump weight, since it merely relocates a portion of it from the engine to a remote energy storage substation. This reduces the maximum flow requirement through the trunk lines which connect from the engine to the energy storage substation. The weight trade which must be evaluated is between the trunk line reduction and the flywheel equipment addition.

3. Flywheel speed reduction versus pump capacity - The weight of a flywheel is greatly effected by the allowable speed reduction, i.e. the minimum speed to which it is allowed to degrade at the end of its duty cycle. If the flywheel is coupled to a variable or fixed delivery pump, the maximum output of the pump reduces in proportion to the flywheel speed, and therefore, the required pump size increases in proportion to the allowable flywheel speed reduction. Each application of a pump-flywheel unit has a combined component trade-off for minimum weight. This trade-off is heavily dependent upon the weight of the flywheel housing evacuation system, mounting, etc. Since these components have not been developed and tested, it is difficult to assign a high degree of confidence to this basic building block in the concept.

4. Flywheel versus accumulator as energy storage substation - The flywheel coupled to a hydraulic pump performs essentially the same function as a hydraulic accumulator in that it supplies flow for peak demands when the maximum steady state supply is exceeded. A basic difference exists in that as an accumulator becomes depleted, the pressure supplied drops and approaches the accumulator precharge levels, while the flywheel driven pump will retain its pressure level until its flow capacity is exceeded.

5. Subsystem weight versus pumping system requirements - There will often be instances where trade-off of hydraulic substation energy storage, when considering a particular subsystem, will yield a weight saving over conventional arrangements. When this occurs it would be essentially the result of reduced trunk line capacity between the engine driven pumps and the subsystem components.

However, when all subsystems are considered it frequently occurs that the engine driven pumping system and trunk lines must be of, or near, the original capacity to support other subsystem functions. Such weight trade-offs cannot be accurately evaluated without taking into account the complete air vehicle systems and determining if all systems could be proportionately reduced by application of hydraulic energy storage substation concepts. In any event, as the number of non-simultaneous peak load subsystems increase, the amount of weight savings which can be allocated to each flywheel substation reduces.

6. Hydraulic actuator area unbalance flow requirement - The concept of hydraulic energy storage substations through flywheels does not assume individual reservoirs, but assumes that fluid is locally pumped from the return lines to the pressure lines. In the case where an actuator has high loads in one direction only and has a large diameter rod to support column loading, the trunk line capacity is essentially established by the volumetric displacement of the rod. Increased flow, through the flywheel substation, from return to pressure does not reduce the trunk line capacity requirement caused by the rod volume.

7. Maximum pump speed and flywheel speed incompatability - Maximum hydraulic pump speeds range from 5000 to 15,000 rpm. Flywheel speeds using optimum design parameters range from 30,000 to 100,000 rpm. In all cases of hydraulic energy storage substations a gear reduction box is required between the flywheel and hydraulic pump.

8. Utility function characteristics and possible "all-mechanical" power extraction - For the cases of utility actuation functions typified by a fixed displacement and time for each cycle, it is possible to assume a flywheel directly coupled through a clutch and gearing to the load. Such a system arrangement could consider the output rate to vary in propertion to the speed decay of the flywheel and the instantaneous gearing ratio existing throughout the cycle. Also it is possible as in a landing gear cycle to use over-center linkage to allow the clutching action to occur while the high inertia members are moving at essentially zero rate and hence requiring essentially zero power dissipation during the slipping portion of engagement.

The system studies from which the above discussions were derived and upon which specific weight reduction were determined follow.

STUDY OF XB-70 ELEVON SUBSYSTEM

Present System - The present elevon system used on the XB-70 consists of 12 individual elevon panels (6 on each side of fuselage center line) each of which is operated by two servo controlled hydraulic linear actuators. (See Figure 1) Alternate actuators in each of these actuator pairs are operated by separate independent hydraulic systems. The failure of one of these hydraulic systems will cut hinge moment (load) capabilities in half but leave low load rate capabilities essentially unaffected.

The basic requirements which this system is designed to meet are as follows:

1. 4,250,000 in-lb hinge moment at a rate of 7 degrees per second per system (or per side, i.e. 6 panels) is the maximum power requirement.

2. Maximum rate equals 28 degrees per second at zero output hinge moment.

3. The outer two panels on each side are deactivated when the wings are folded during the high speed portion of the mission profile.

A plan view of the general layout of the system is shown in Figure 1. The mean distance from the power source (secondary power bays) to the elevons is approximately 90 feet and the mean tubing size is 1-1/8 inch diameter.

The measured demands of the XB-70 flight control system show the following hydraulic system (one system) requirements:

Fuel pump drive hydraulic motors (steady state)	- 16 GPM
Elevons - Low Hinge Moment (low pressure) at 25°/sec (Maximum average)	- 124.6 GPM
Elevons - Continuous motion, High hinge moment 3°/sec	- 15 GPM
FACS servos (yaw, pitch, roll) Master Actuators	- 6 GPM
System Leakage (continuous)	- 9.6 GPM
Wing Fold	- 33.4 GPM

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CHECKED BY:	XB-70 ELEVON HYDRAULIC	REPORT NO. NA-65-825-1
DATE: 19 JAN 65	SYSTEM SCHEMATIC	MODEL NO.

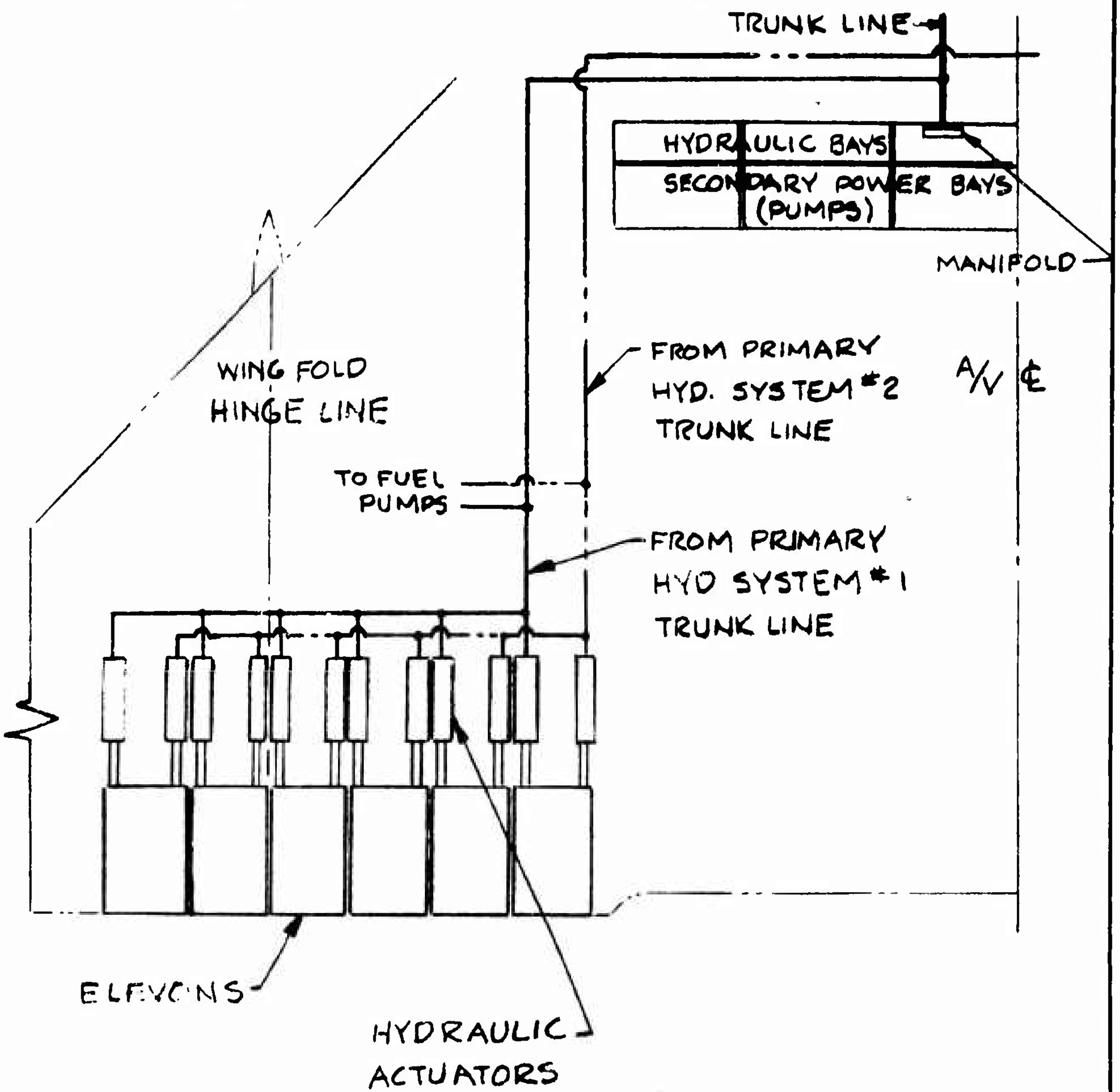


FIG 1

Hydraulic Energy Storage Substation

The substation is integrated into the system as shown in Figure 2 and consists of:

1. A fixed angle, variable displacement motor-pump unit, 1.25 in^3/REV. maximum displacement, which supplied 30 GPM at 6000 RPM and which will wind down to 22.5 GPM at 4500 RPM is selected as the flywheel power transmission device. Input flow will be from a 4000 PSI system with line loss set at 1000 PSI. Effective motoring pressure will then be 3000 PSI. One unit in each wing will supply a single system. (4 will be required for the total A/V two system requirements.)

   Windage loss of the motor is estimated by extrapolation of measured torque values of 5 in^3/REV and 2.5 in^3/REV displacement Vickers pumping units. This value is 210 in-lbs and by conversion:

   $$HP = \frac{(210 \text{ in-lbs})\ (6000 \text{ RPM})}{63025} = 20 \text{ HP/UNIT}$$

2. A gearbox which will transmit 60 horsepower maximum and have a motor to flywheel gear ratio of 6000 RPM to 40,000 RPM.

   Windage losses of the gearbox are estimated at 8 percent of total transmitted peak HP.

   $$HP \text{ loss} = .08\ (60) = 4.8 \text{ HP/UNIT}$$

3. A flywheel enclosed in an evacuated case and sized as follows is used in the energy storing device

   a. Flight Control Peak Requirements

Elevons	124 GPM
FACS & Master Act.	6 GPM
Sys. Leakage	9.6 GPM
	139.6 GPM/SYS

   b. Supply system

Substation 22.5 GPM/Unit	45 GPM
*Engine Driven Pumps	94 GPM
	139 GPM/SYS

*NOTE:	Flight Control Requirements	94 GPM
	Fuel Pump Drive	16 GPM
		110 GPM/SYS.

   110 GPM/SYS Required at 61% engine RPM

   61% (180) = 110 GPM = (3 pumps at 60 GPM each).

PREPARED BY: HELSLEY	NORTH AMERICAN AVIATION, INC.	PAGE NO. 7 OF
CHECKED BY:	PROPOSED XB-70 ELEVON	REPORT NO. NA-65-825-1
DATE: 24 JAN 66	HYDRAULIC SUBSTATION SYSTEM	MODEL NO.

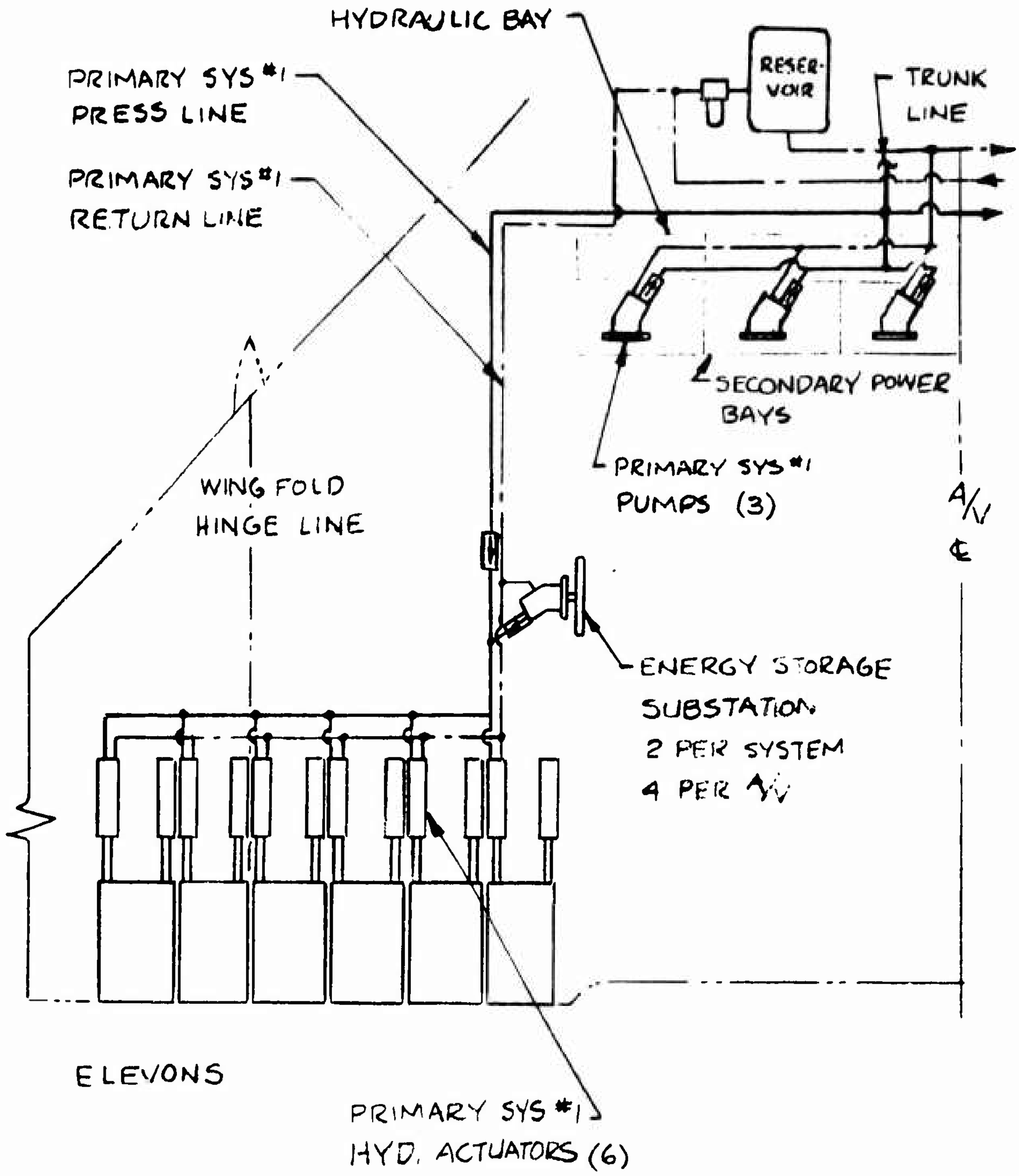


FIGURE 2

3. Continued

c. Flywheel sizing

$$\text{HP system} = \frac{(30\ \text{GPM})\ (2600\ \text{PSI})}{1714} = 45.5$$

HP gearbox windage	=	4.8
HP pump windage	=	20.0
HP flywheel windage (See report NA65-825 - Evacuated Flywheel)	=	1.5
TOTAL	=	71.8 HP

Taken for 12 SEC period at 50% of time

HP SEC = 71.8 x 6 = 430

From Figure 4 NA65-825

Flywheel radius	-	5.0 IN
weight	-	13.0 LBS
RPM	=	43200.0

Systems Evaluation

Steady state flow requirements are as follows:

a. Fuel Pump Drive 16 GPM

b. Elevons 15 GPM

c. FACS & Master Act. 6 GPM

d. System Leakage 9 GPM

e. Substation Windage

$$\left(\frac{26.3\ \text{HP} \times 1714}{3000}\right)^2 = 30\ \text{GPM}$$

f. Total steady state flow 76 GPM

Peak requirements are as follows: (At Substation)

a. Elevons 124 GPM

b. FACS & Master Act. & Leakage 15 GPM

139 GPM

As substations are capable of 45 GPM output peak line flow from engine pumps to wings is equal to 94 GPM. As soon as steady state conditions are restored to 76 GPM a minimum of 18.0 GPM will be available for substations windup.

Wt. Comparisons

ITEM (ENG. STARTING NOT CONSIDERED)	WEIGHT ESTIMATE SUB-STATION CONCEPT	PRESENT
Engine Driven Pumps 3 required	180	300
Lines 94 GPM vs 154 GPM	115.2	160.5
Fittings	70	100
Supports	20	40
Motor Pump 2 (30 GPM at 6000 RPM)	60	
Gear Box 2 units required	24	
Subsystem Installation	20	
Flywheels Incl. Housing and Support 2 required	46	
	534.2	600.5

Mechanical - Hydraulic Energy Storage Substation

The proposed system is shown schematically in Figure 3. This system utilizes six power hinges per side, one for each elevon panel. There are, however, only three energy storage substations. The three substations utilize output shear shafts so that in event of failure of any one substation the other two can shear the third substation's shaft and continue elevon operation unimpaired. In contrast to the hydraulic system the energy storage system will loose only 1/3 of its hinge moment capabilities and none of its rate capabilities in the event of failure of one power supply system or of the energy storage substation itself. It will be noted that all the energy storage substations are located inboard of the wing fold line. This is necessitated by the fact that the outboard elevons must be deactivated after wing folding.

The basic requirements which the XB-70 elevon powering system must meet are tabulated in the following list:

PREPARED BY: HELSLEY	NORTH AMERICAN AVIATION, INC.	PAGE NO. 10 OF
CHECKED BY:	PROPOSED XB-70 ELEVON	REPORT NO. NA-65-825-1
DATE: 19 JAN 65	MECHANICAL SYSTEM	MODEL NO.

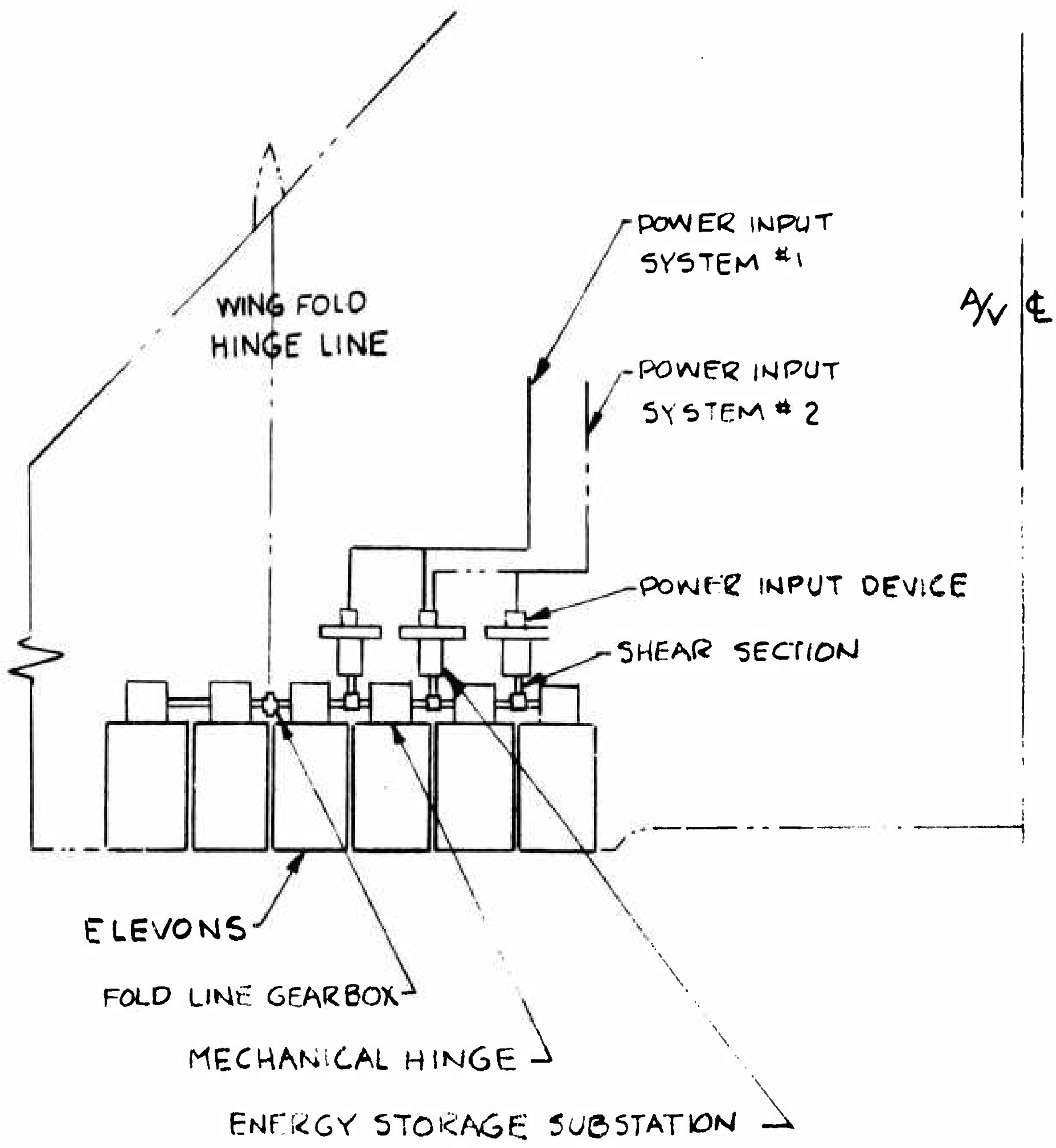


FIG 3

(HM)	Max. Hinge Moment (at surface)	=	4,250,000 in-lb
	Max. Hinge Moment per elevon	=	708,333 in-lb
(ω_p)	Max. Power Rate	=	7°/sec
		=	1.16 RPM
		=	.1215 Rad/sec
(ω_m)	Max. Rate	=	28°/sec = 4.54 RPM
(θ)	Max. Deflection (amplitude)	=	± 28°
		=	.486 Rad.

It is assumed that in the worst condition, the elevon powering system will have to meet is a sinusoidal power output duty cycle in which the elevons go through a maximum excursion of ± 28 degrees. It is further assumed that the peak power requirement occurs at 4,250,000 in-lb and 7°/sec. The resulting cycle is as shown in Figure 4. It is obvious that the velocity would call for infinite acceleration. However, since the inertia of the elevons is very small relative to the loads involved, the adoption of this simplifying assumption does not lead to significant error.

On this basis it can be said that:

$$P_a = \frac{P_m}{\sqrt{2}}$$

$$= .707\ P_m$$

Where:

P_a = average power delivered (HP)

P_m = max. power delivered (HP)

And

$$P = \frac{HM \times \omega}{63025}$$

Where:

HM = Hinge moment in-lb

ω = Angular velocity (RPM)

Also it can be said that:

$$HMA = \frac{HM_{max}}{2}$$

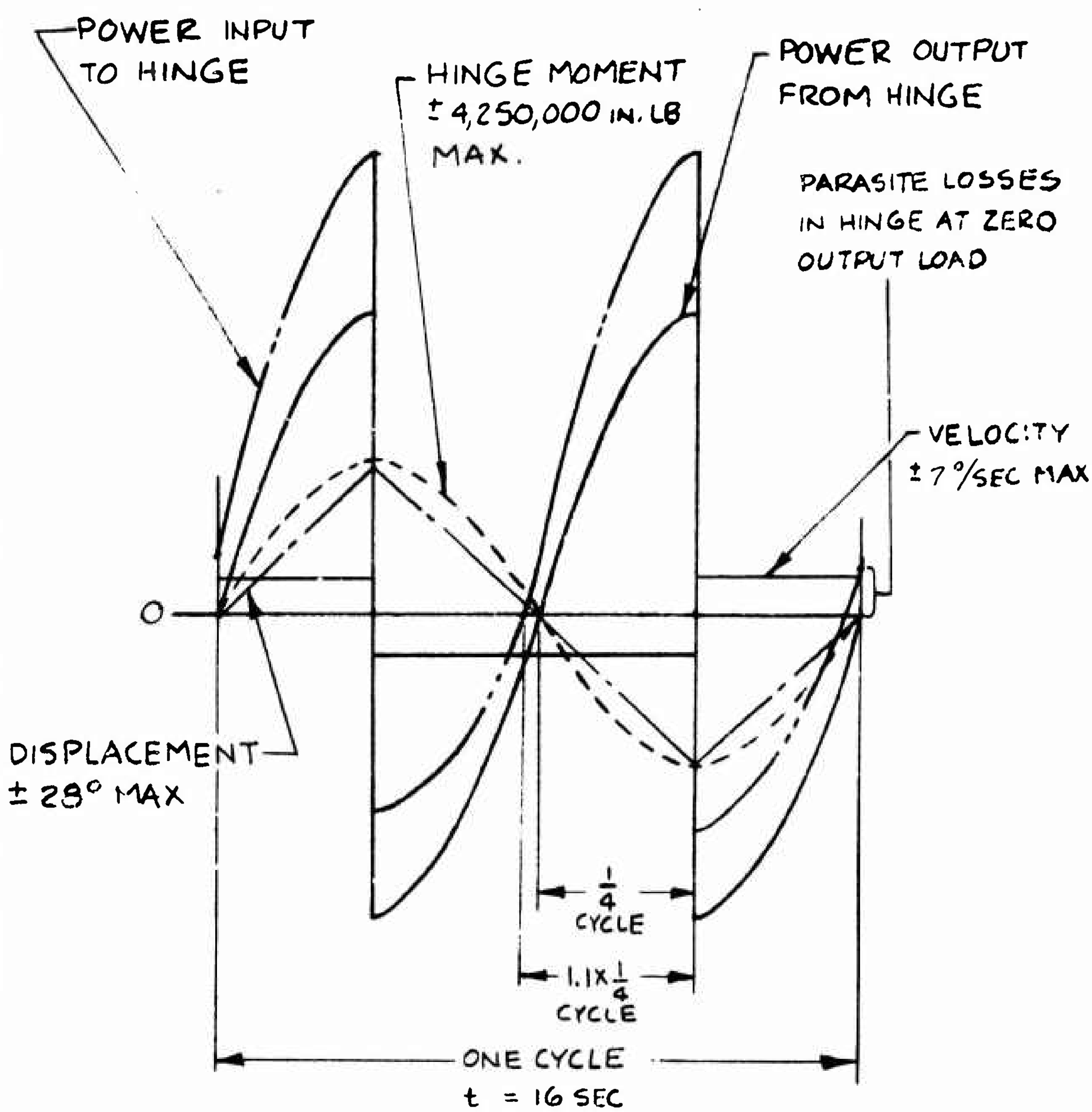

POWER INPUT TO HINGE
HINGE MOMENT ± 4,250,000 IN. LB MAX.
POWER OUTPUT FROM HINGE
PARASITE LOSSES IN HINGE AT ZERO OUTPUT LOAD
VELOCITY ± 7 °/SEC MAX
0
DISPLACEMENT ± 28° MAX
1/4 CYCLE
1.1X 1/4 CYCLE
ONE CYCLE t = 16 SEC

FIG 4

Therefore, in the case of the XB-70 elevon system, the input hinge moment (HM_{Ai}) to the mechanical hinge (Figure 2) assuming the hinge is 65 percent efficient will be

$$HM_{Ai} = \frac{.707 \times 708,333}{.65}$$

$$= 770,000 \text{ in-lb.}$$

during the opposing load portion of the elevon motion.

During the aiding load portion of the motion, assuming the reverse power flow efficiency is the same as normal flow, the aiding hinge moment is:

$$HM_{Ai} = .65 \times .707 \times 708,333$$

$$= 325,000 \text{ in-lb}$$

It will be noted from further examination of Figure 3 that there is a parasite loss associated with the mechanical hinge which means that, when the output power requirement goes to zero, there is still a significant input power requirement to overcome the bearing and lubricant threshing, windage, and friction losses. The magnitude of these losses vary depending upon the design and gear reduction characteristics of the power hinge. However, it can be safely said that they will seldom exceed 10 percent of the maximum rated power output of the hinge.

Based upon this fact the simplifying assumption made to approximate these losses is to consider that the average power input to the hinge during the opposing load portion of the cycle occurs during 110 percent of the half cycle and conversely that average power recovery during the aiding load portion occurs during only 90 percent of its half cycle. This is shown graphically in Figure 3.

Based upon this modification therefore the hinge moments just determined would be modified as follows:

$$HM_{Ai} \text{ (opposing)} = 770,000 \times 1.1 = 848,000 \text{ in-lb}$$

$$HM_{Ai} \text{ (aiding)} = 325,000 \times 0.9 = 292,000 \text{ in-lb}$$

The net energy which must be supplied at each power hinge input during each cycle if the storage system is not to be progressively depleted is:

$$HM_{Ai} \text{ net} = (848,000 - 292,000)$$

$$= 556,000 \text{ in-lb/cycle/elevon panel}$$

Therefore:

$$P_A = \frac{556,000 \times 1.16}{63,025} = 10.20 \text{ HP/panel}$$

or for the complete system of 12 panels:

$$P_A = 122.4 \text{ HP}$$

Based upon the preceeding data the peak torque (T_S) demand from the substation, assuming 98 percent transmission efficiency from the substation to the mechanical hinge and a 5000 to 1 speed reduction ratio for the mechanical hinge, would be:

$$T_S = \frac{708,333 \times 2}{(.65) \times (.98) \times (5000)} = 445 \text{ in-lb/substation}$$

To meet the maximum surface rate requirement of 28°/sec the substation output speed would be:

$$\omega_S = \omega_{surface} \text{ X gear ratio}$$

$$= 4.54 \text{ RPM} \times 5000$$

$$= 22,600 \text{ RPM}$$

The size of the flywheel used in the substation will be determined by the magnitude of the excess energy demand during that portion of the cycle where energy demand exceeds energy supplied. For this system, as has already been shown, the power supplied to prevent rundown will be P_A = 10.2 HP/panel or 20.4 HP/substation. This can be converted to:

$$P_A = 20.4 \frac{\text{HP}}{\text{substation}} \times 6000 \frac{\text{in-lb.}}{\text{sec HP}} = 134,500 \frac{\text{in-lb}}{\text{sec-substation}}$$

The 1/2 cycle during which opposing loads exist represents that portion of the cycle where energy demand exceeds energy supply. The energy balance during this period is as follows:

Energy demand work $= HM_{Ai} \times$ distance (20)

$$= 848,000 \frac{\text{in-lb}}{\text{elevon}} \times 2 \text{ elevons} \times .972$$

$$= 1,643,000 \text{ in-lb}$$

Energy supplied work $= P_A \times 1.1\ t$

Where: t = time in seconds for 1/2 cycle

$$t = \frac{2 \times .486 \text{ rad/1-2 cycle}}{.1215 \text{ rad/sec}}$$

$$t = 8 \text{ seconds}$$

Therefore:

Energy supplied = 134,000 x 8.8 = 1,178,000 in-lb

Net energy required = 1,643,000 - 1,178,000

= 465,000 in-lb

Assuming an allowable flywheel speed reduction of 10 percent, the total energy storage capability (from Figure 4, NA65-825) should be approximately 2,450,000 in-lb.

A mechanical energy storage substation which should meet these requirements is shown in Figure 5. The flywheel is integrated with the input toroid of the mechanical servo, therefore it is assumed that a lower fatigue strength material will be used to provide leeway for selecting the material with the most desirable bearing properties at the roller-toroid contact point. It is assumed that the material, as a flywheel, has an endurance limit of 100,000 psi. Using the previously determined total energy requirement the size, shape, and weight of the flywheel can be closely approximated from page 422, Appendix A of NA65-825. This shows that an 8 inch diameter, .030 inch tip thickness, 9.536 lb flywheel has a kinetic energy slightly in excess of that required.

Lumping this weight in with that required for a flywheel housing and a properly sized mechanical servo gives a total energy storage substation weight of 60 lbs. Additional data on the expected performance of the energy storage substation including an expected 94 percent transmission efficiency at 42.5 HP (rated output) is shown in Figure 5.

As has been previously pointed out the portion of the total power supplied to the mechanical hinge which must be drawn from the main system to prevent run down is 20.4 HP. Allowing for the efficiencies of the intermediate shafting and the energy storage substation the power which must be supplied to the energy storage substation input is as follows:

$$P_1 = \frac{20.4}{(.94)(.98)} = 22.1 \text{ HP}$$

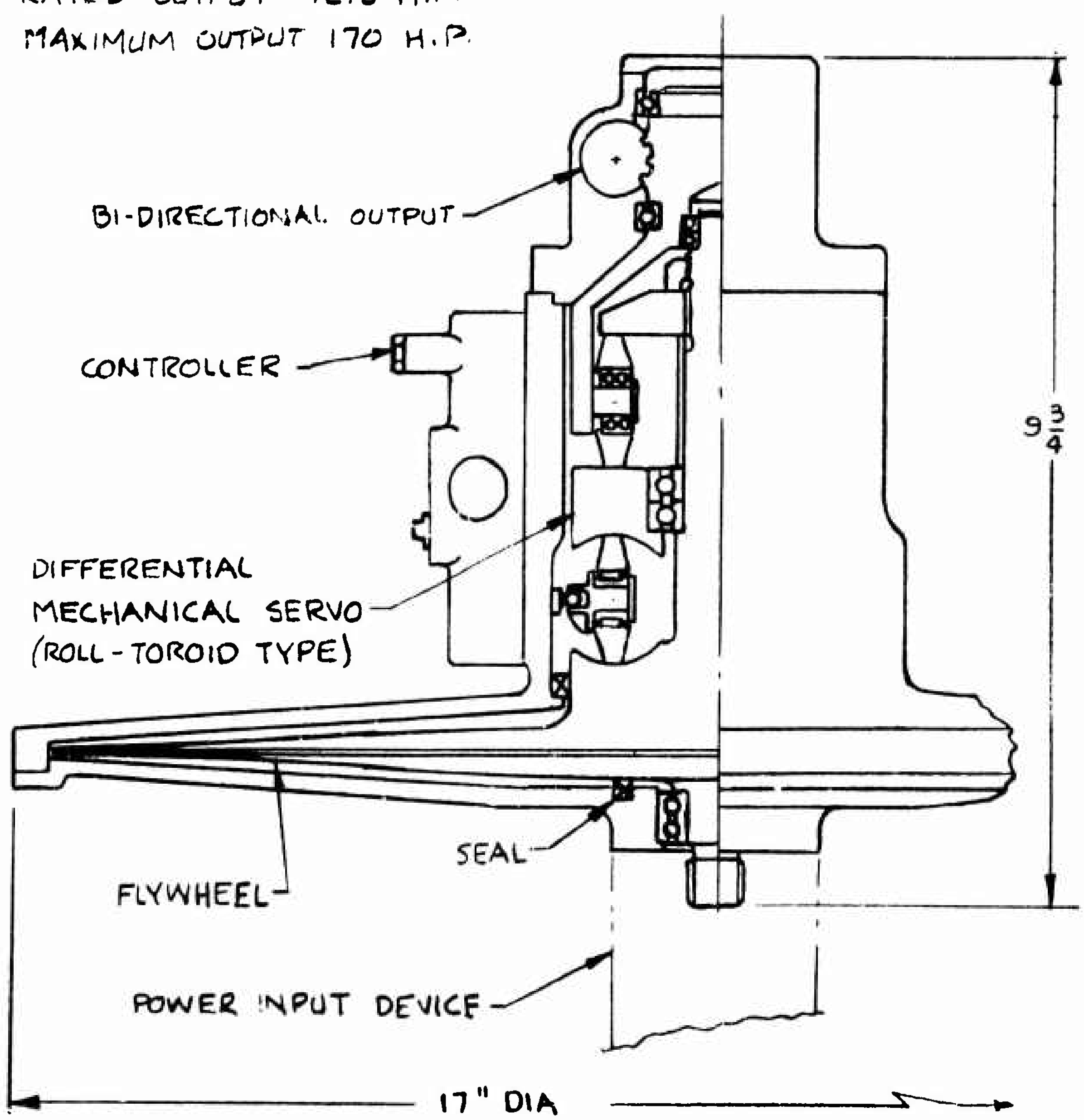
The power input device can be pneumatic, hot gas, mechanical, electrical, or hydraulic. For the purposes of this study a hydraulic power input device will be assumed. Therefore based upon a 4000 psi system (3000 psi differential pressure available across the device) the input flow required will be:

$$Q = \frac{1714 \times P}{psi}$$

$$Q = \frac{1714 \times 22.1}{3000} = 12.6 \text{ GPM}$$

FIG 5

- INPUT SPEED = 33,000 RPM
- MAX ROLLER-TOROID SURFACE VELOCITY = 5000 IN/SEC
- TRACTIVE EFFORT = 150 LB/ROLL = 450 LB/SERVO
- PEAK OUTPUT TORQUE 550 IN-LB
- OUT-PUT SPEED = ± 22,600 RPM (AT WORM)
- WEIGHT = 60 LB
- TRANSMISSION EFFICIENCY = 94%
- RATED OUTPUT 42.5 H.P.
- MAXIMUM OUTPUT 170 H.P.



To meet this demand a hydraulic motor with 0.5 in3/rev. displacement operating at 7000 rpm will be required. The approximate windage losses for a unit this size operating at this speed will be about 3.9 HP or 2.2 GPM.

Therefore the flow input required per substation will be:

$$Q = 14.8 \text{ GPM}$$

and the flow per system or side:

$$Q_s = 3Q = 44.4 \text{ GPM}$$

and the total flow for the six substations on the air vehicle is:

$$Q_t = 2Q_s = 88.8 \text{ GPM}$$

Based upon the flow per system (Q_s) determined above, the engine mounted pumps would be sized as shown in the following tabulation:

Total simultaneous flow requirements for primary hydraulic system.

Fuel Pump Drive	16 GPM
FACS & Master Actuator	6 GPM
System Leakage	9 GPM
Elevon Substations	44.4 GPM
	75.4 GPM

A summary of the weights of the components making up this system and its total weight is shown in Table IV. This table also compares the weight of this system with the other two elevon system approaches studies, i.e, the base line (present) hydraulic system and the all hydraulic energy storage substation system.

TABLE I

COMPARATIVE SYSTEM WEIGHTS (LBS)

COMPONENT DESCRIPTION	PRESENT XB-70 ELEVON SUBSYSTEM	HYDRAULIC ENERGY STORAGE SUBSTATION SYSTEM	MECH HYDRAULIC ENERGY STORAGE SUBSTATION SYSTEM
Engine Driven Pumps	300 (3-154 GPM)	180 (3-94 GPM)	150 (3-74.5 GPM)
Hydraulic Lines	160.6	115.2	100
Hydraulic Fittings	100	70	60
Line Supports	40	20	18
Manifold Filters, Ripple Dampers, etc.	624.9	463.5	371
Hydraulic Actuators	780	780	-
Substation Powering Device	-	60 (2 - 30 GPM - 6000 RPM)	24 (3 - 14 GPM - 7000 RPM)
Auxiliary Gearboxes	-	24	10
Substation incl. Flywheel	-	46 (2 required)	180 (3 required)
Substation Instal. Brackets, etc.	-	80	60
Mechanical Hinges	-	-	540
Connecting Shafts	-	-	17
Heat Exchangers	-	-	9
TOTALS	2005.5 lbs	1848.7 lbs	1539 lbs
Weight Savings	None	156.8	466.5

STUDY OF XB-70 LANDING GEAR SUBSYSTEM

Preface

This study attempts to establish system concepts and weight comparisons between the first three "Possible Configurations" as listed in the contract for an intermittent duty type function.

Discussion

Energy stored in the form of motion can be converted into hydraulic energy only through a pump. Since pump size is dictated by rate requirements it follows that for a given operating pressure, the pumping capacity of the sub-stations plus the engine driven pumps (configuration 2 of the contract) must be at least as large as the engine-driven pumps in the standard system (configuration 1 of the contract). Therefore, any added weight of flywheel, gearbox, filters, etc., required for the energy storage substation configuration must be off-set by savings in weight of the trunk lines and engine-driven pumping system. The large air vehicle with long, large diameter trunk lines appears to offer the best opportunity for this stored energy concept, therefore, the landing gear requirements of the XB-70 are used in this study.

One approach is to install one flywheel, geared to a motor-pump, in each wheel well to supply as much as possible of the energy used in that wheel well, while requiring the engine-driven pumps and trunk lines to supply only enough fluid to charge up the flywheels prior to use and to make up any deficiency in fluid resulting from unbalanced actuators and fluid compressability. This is investigated as configuration 2 and conforms to configuration 2 of the contract.

The XB-70 gear system is designed to minimize installed weight, to be as simple as possible, and to have maximum reliability. It is not economical in terms of total quantity of energy used. This wasted energy which represents a negligible quantity of fuel, results from the use of flow regulators to maintain constant rate in landing gear operations, regardless of load variation.

In a stored energy system this waste may result in a noticeable increase in weight of storage equipment, therefore, a second system is investigated. Configuration 3 represents a system in which a rotary mechanical hinge, driven directly by a flywheel, powers the individual operation. Due to acceleration rate control inherent in over-center linkages, this system minimizes wasted energy while providing simplicity in speed control. However, since subsystems which power non-simultaneous functions, demand a continuous flow of energy to keep their flywheels wound up, it follows that the weight saving in engine pumps and trunk lines tends to disappear as the number of subsystems increases.

CONFIGURATION #1

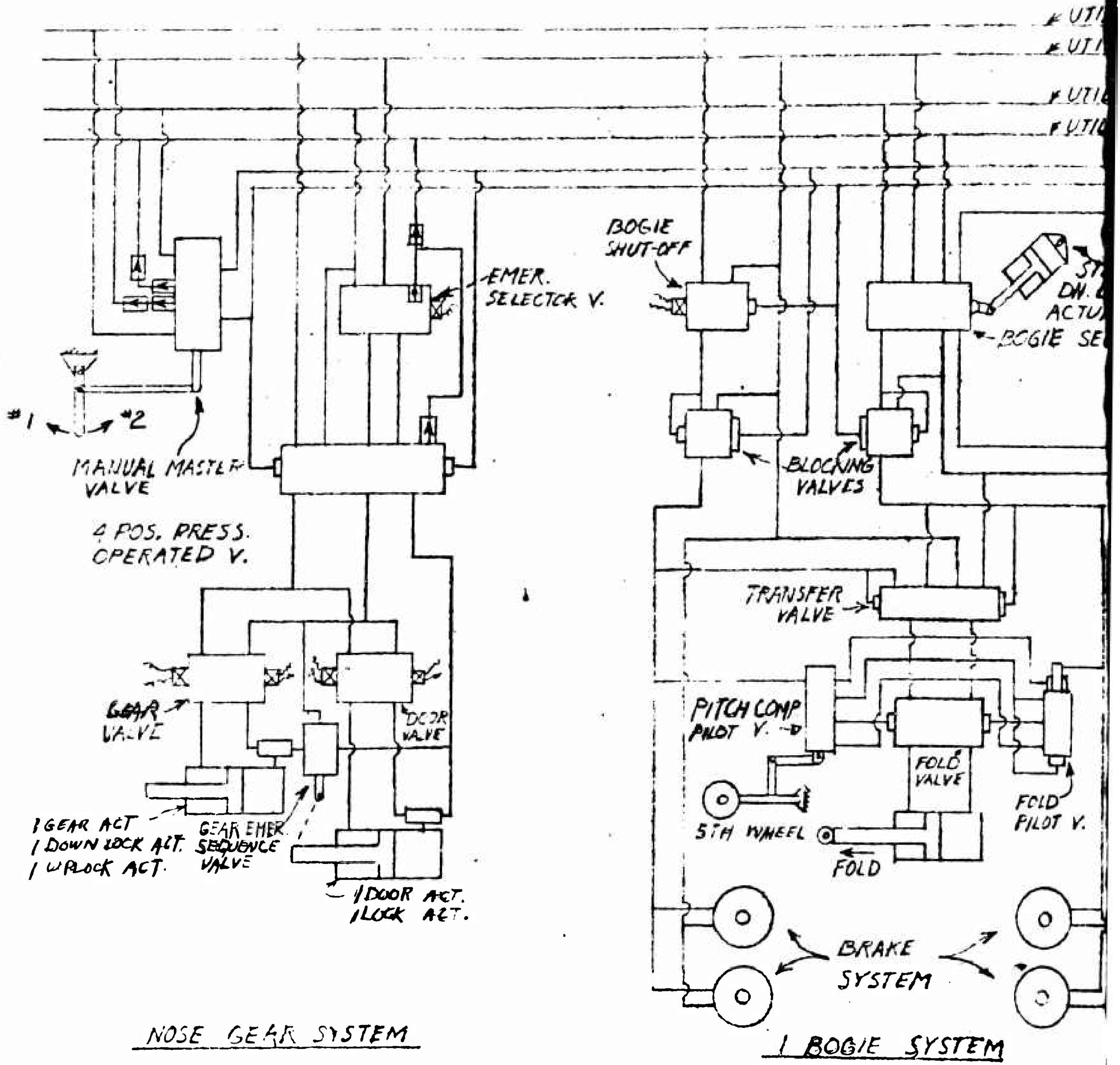
Description

Primary Power - Jet Engine
Secondary Power - Hydraulic Pumps
Control - Hydraulic Valves
Actuator - Linear Hydraulic

This configuration is that used in the B-70 air vehicle as well as most other aircraft. It will be used as the standard against which other configurations will be evaluated. To simplify evaluation, B-70 landing gear system will be used but the pumping and distribution system will be resized to support only the landing gear system. Other hydraulic loads tend to be additive and will be omitted.

In this configuration two hydraulic systems divide the gear functions for normal operation and either system can power all functions at reduced rate for emergency operation. A three line system with shuttle valve at each actuator is used.

Figure 6 is a simplified schematic of this system. Table II shows the power requirements for configuration 1.


UTI
UTI
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UTI
BOGIE SHUT-OFF
EMER. SELECTOR V.
BOGIE SE
ACTU
#1
#2
MANUAL MASTER VALVE
BLOCKING VALVES
4 POS. PRESS. OPERATED V.
TRANSFER VALVE
GEAR VALVE
DOOR VALVE
PITCH COMP PILOT V.
FOLD VALVE
FOLD PILOT V.
1 GEAR ACT
1 DOWN LOCK ACT.
1 UPLOCK ACT.
GEAR EMER. SEQUENCE VALVE
5TH WHEEL
FOLD
1 DOOR ACT.
1 LOCK ACT.
BRAKE SYSTEM
NOSE GEAR SYSTEM
1 BOGIE SYSTEM

FIGURE 6

SIMPLIFIED SCHEMATIC B-70 AIRCRAFT

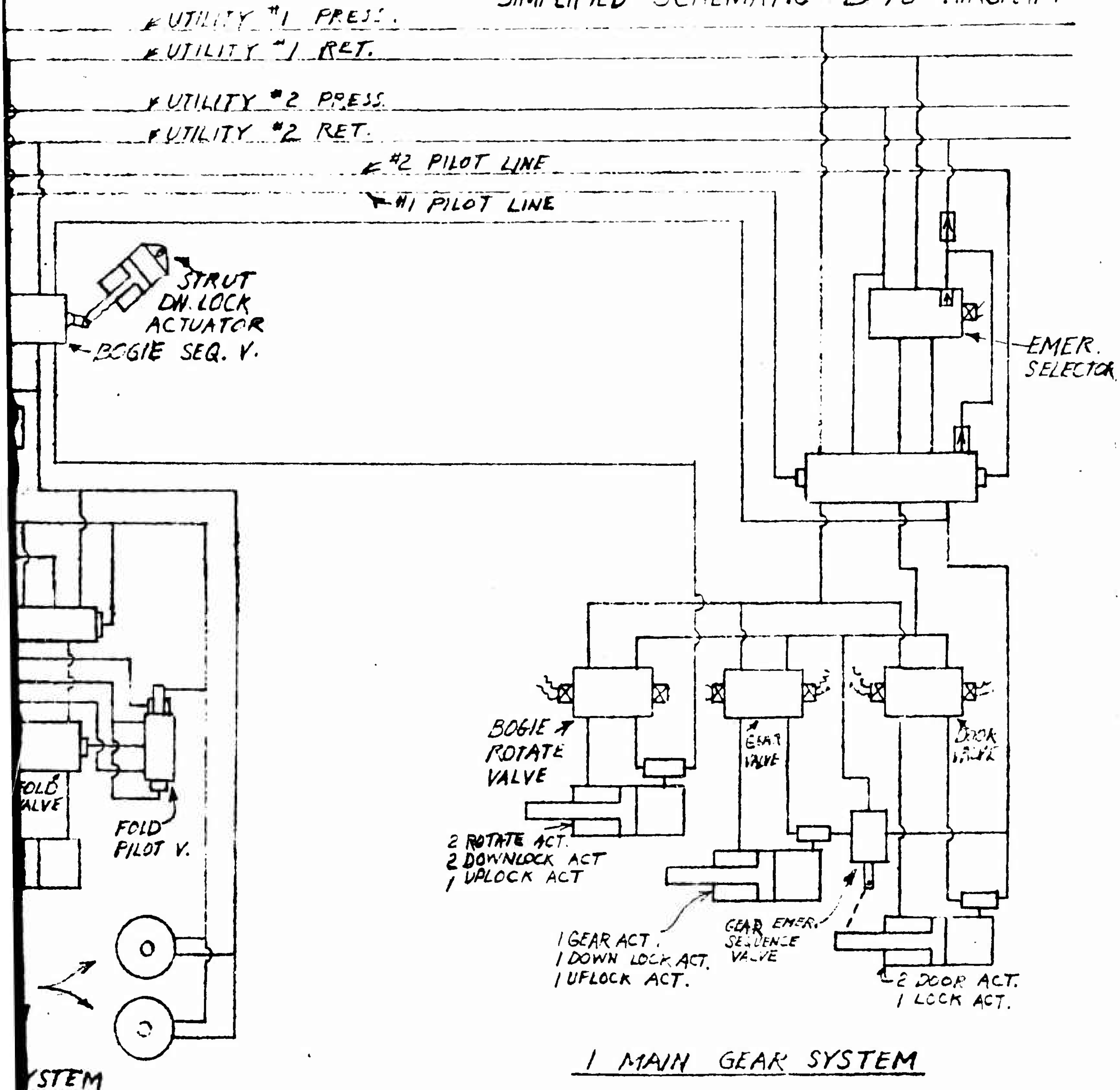

TABLE II

LANDING GEAR POWER REQUIREMENTS

FUNCTION	Operating Time Sec	Flow Req. GPM
Nose Gear		
Door Open	4.5	5.1
Door close	3.0	5.1
Gear Extend.	15.9	5.5
Gear Retract	11.3	4.5
Steering	5.0 per application	7.0
Main Gear		
Door Open	4.5	11.3
Door Close	3.0	12.3
Gear Extend	12.5	12.5
Gear Retract	7.4	15.1
Bogie Rotate	5.5	5.5
Bogie Unfold	5.5	19.7
Bogie Fold	5.5	24.4
Brakes	2.0 per application	6.4 (Ave) 10.8 (Peak

Combinations of Flow Required:

Normal operations - System 1

1. Nose Door Open + Main Door Open = 5.1 + 2(11.3)= 27.7 GPM
2. Nose Gear Extend + Main Gear Extend = 5.5+ 2(12.5)=30.5 GPM
3. Steering + Brakes = 7 + 2(10.8) = 28.6 GPM

Normal operations - System 2

1. Bogie Fold = 2(24.4) = 48.8 GPM
2. Brakes = 2(10.8) = 21.6 GPM

Engine Pump Size

Pump efficiency at 400 F = 86% (assumed)
Engine RPM at gear extention = 80% (assumed)

System 1 $\dfrac{30.5}{80\% \times 86\%} = 44.4$ GPM

System 2 $\dfrac{48.4}{86\%} = 56.4$ GPM

Two PV062 Pumps at 7000 RPM = 56.4 GPM

COMPARISON WEIGHT OF CONFIGURATION 1

Weight of items which will be changed in other configurations:

Eng. Pump, lb.	(2 PV062)			56
Filter	50 GPM			18
Trunk Lines & Fluid:				
Main Gr	116 x 1-1/8 OD		=	96
Nose Gr	112 x 1/2 OD		=	20
Line supports, etc.				116
				306

A/V Weight , Total = 612 LB

CONFIGURATION #2

DESCRIPTION

Primary Power - Jet Engine
Secondary Power - Hydraulic Pump
Energy Storage Subsystem - Hydraulic Motor-Pump-Flywheel
Control - Hydraulic Valve
Actuator - Linear Hydraulic

This configuration consists of a flywheel geared to a motor-pump, in each wheel well, to supply as much as possible of the energy used in that area. The engine driven pumps and trunk lines are sized to charge up the flywheels prior to use, and to supply sufficient flow to make up the suction requirements of the wheel well mounted pumps resulting from fluid compressibility and differential displacement of unbalanced actuators.

To be comparable to the B-70 arrangement the bogie is folded by one system and other gear operations are powered by the opposite system. The flywheels will be sized for a 10% speed reduction and in emergency use with one system out of action, the extra demand will be supplied at a lower rate as the flywheel slows below 90% speed.

Table III shows the flow requirements needed to maintain the inlet fluid supply of the flywheel powered pumps when system actuators are extending.

Figure 7 shows the schematic of a flywheel stored energy system. Additional functions, such as wheel door, bogie rotation, etc., are provided by addition of another valve, plumbed in parallel to the first one. When any valve is energized to an actuation position, the drop in system pressure switches the motor-pump from the motor mode to the pumping mode. When the function is completed the pressure rise will return the motor-pump to the motor-mode.

Emergency extension is provided by the other system, supplying pressure through emergency lines and shuttle valves at the actuators. Return flow is routed to its proper system by the emergency selector valve.

Reliability is therefore, that of two separate systems down to the shuttle valve on each actuator, as in the basic B-70 prior to addition of the completely non-electrical gear extension system.

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CHECKED BY:	FIGURE 7	REPORT NO. NA-65-825-1
DATE:	SCHEMATIC - ENERGY STORAGE SUBSYSTEM - CONFIGURATION 2.	MODEL NO.

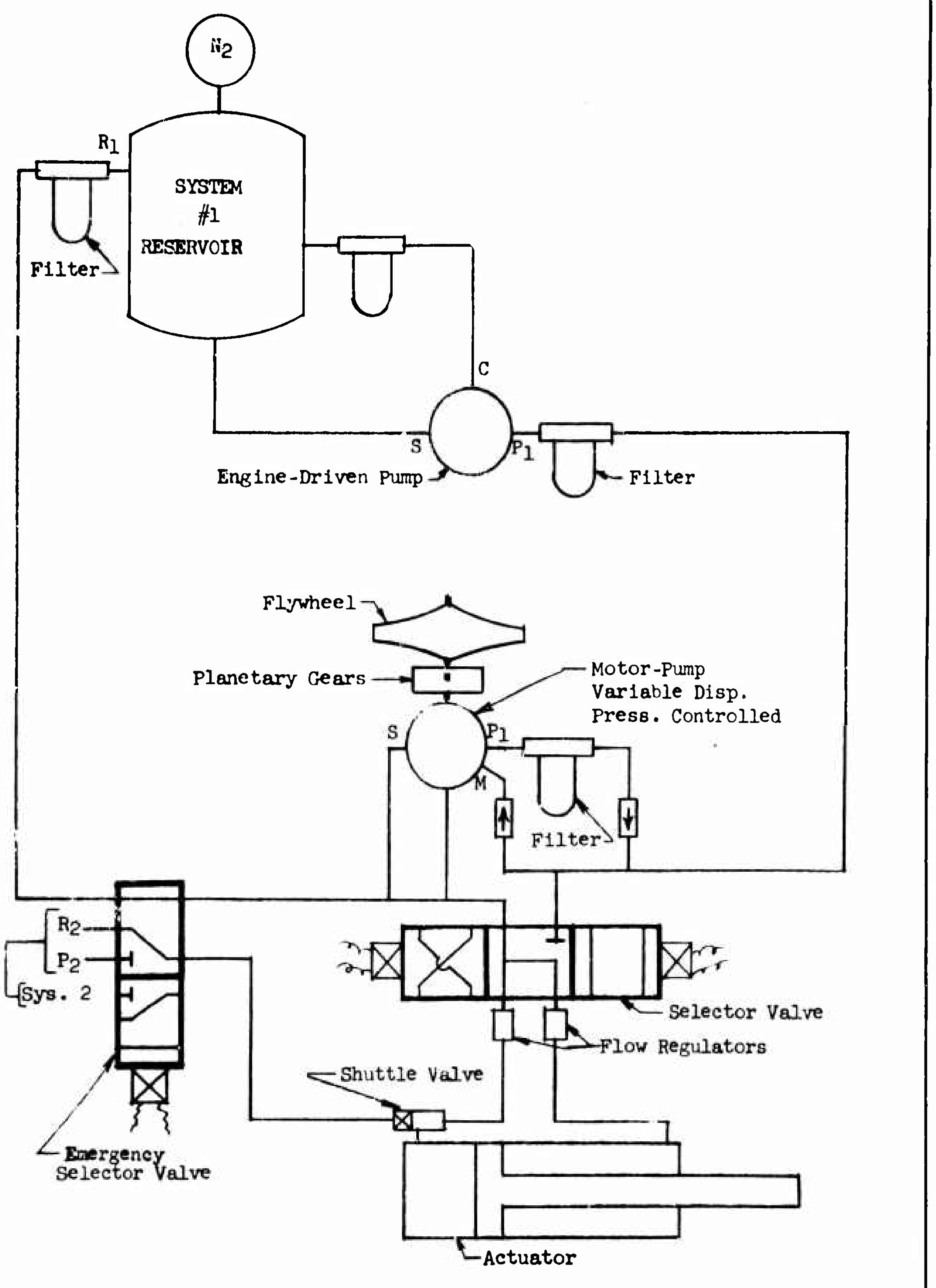


TABLE III

ENGINE-DRIVEN PUMP SIZED TO SUPPLY INLET FLOW FOR FLYWHEEL PUMPS

FUNCTION	DISPLACEMENT IN Cu. In.	DISPLACEMENT Out Cu. In	OPER. Time Sec.	Differential Flow Req. from Eng. Pump Cu In/Sec	Differential Flow Req. from Eng. Pump GPM	TOTAL Flow Req. GPM
Nose Gear						
Door Open	87.1	55.8	4.5	31	1.8	5.1
Door Close	55.8	87.1	3.0	-	-	5.1
Gear Ext.	337.	194.	15.9	143	2.3	5.5
Gear Ret.	194.	337.	11.3	-	-	4.5
Steering	78.2	78.2	5.0 per applica.	-	-	7.0
Main Gear						
Door Open	192.	134.	4.5	58	3.4	11.3
Door Close	134.	192.	3.0	-		12.3
Gear Ext.	594	424	12.5	170	3.6	12.5
Gear Ret.	424	594	7.4	-		15.1
Bogie Rot. DN	126	112	5.5	14	.7	5.5
" " UP	112	126	5.5	- 14	- .7	5.5
" "Unfold	411	517	5.5	-	-	19.7
" " Fold	517	411	5.5	105	5.0	24.4
Brakes	24	12	2.0 per applica.	12	3.1	6.4 Ave 10.8 Peak

COMBINATIONS OF DIFFERENTIAL FLOWS REQUIRED FROM ENGINE-DRIVEN PUMPS

Normal or Emergency (Split or single) System Operation

1. Nose Door + Main Door Open = 1.8 + 2(3.4) = 8.6 GPM
2. Nose Gear + Main Gr Ext. 2.3 + 2(3.6) = 9.5 GPM
3. Nose Gr Ext. + Bogie Rot. + Unfold = 2.3 + 2(.7) = 3.7 GPM
4. Bogie Fold = 2(5.0) =10 GPM
5. Steering + Brakes = 2(3.1) = 6.2 GPM

Maximum pump size required for gear extension with engine at 80% RPM:

$$\frac{8.5}{80\% \times 86\%} = 13.8 \text{ GPM}$$

Maximum pump size required for gear retraction with engine at 100% RPM:

$$\frac{10}{86\%} = 11.65 \text{ GPM}$$

The engine driven-pump will be a PV024 producing 13.8 GPM at 8800 RPM.

TABLE IV

ENERGY STORAGE SUBSYSTEM PUMP & FLYWHEEL SIZING

Energy Storage Equipment
Sized for Min. Weight
Eng. Pump & Trunk Lines

Engine-Driven Pump Output

Gross: 13.8 GPM at 100% RPM
Net: 11.9 GPM at 100% RPM
9.5 GPM at 80% RPM

FUNCTION	Flow Req. 3700 psi	From Eng. Pump		Energy Req. From Flywheel Pump				Load Duration	Stored Energy Ft-Lbs	
		GPM		GPM		HP				
	GPM	RET	EXT	RET	EXT	RET	EXT	SEC	RET	EXT
Nose Gear										
Door Open	5.1	1.8	1.8	3.3	3.3	7.1	7.1	4.5	17,600	17,600
Door Close	5.1	-	-	5.1	5.1	11.0	11.0	3.0	18,100	18,100
Gear Ext.	5.5	-	2.3	-	3.2	-	6.9	15.9	-	60,300
Gear Ret.	4.5	-	-	4.5	-	9.7	-	11.3	60,300	-
								Total	96,000	96,000
Steering										
No Brakes	7.0 peak	4.7	2.3	2.3	4.7	5.0	10.3	5.0 per appli-	13,700	28,000
With Brakes	7.0 peak	-	-	7.0	7.0	15.1	15.1	cation	41,500	41,500
Main Gear										
Door Open	11.3	5.0	3.8	6.3	7.5	13.6	16.2	4.5	33,600	40,100
Door Closed	12.3	5.9	4.7	6.4	7.6	13.8	16.4	3.0	22,800	27,100
Gear Ext.	12.5	-	3.6	-	8.9	-	19.2	12.5	-	132,000
Gear Ret.	15.1	5.9	-	9.2	-	19.9	-	7.4	81,200	-
Bogie Rotate	5.5	5.5	3.6	-	1.9	-	4.1	5.5	-	12,400
								Total	137,600	211,600
Bogie Unfold	19.7	-	4.7	-	15.0	-	32.4	5.5	-	98,000
Bogie Fold	24.4	5.9	-	18.5	-	40.0	-	5.5	121,000	-
Brakes										
No Steering	6.4 Ave.	4.8	3.6	1.6	2.8	3.5	6.0	2 to 30	55,000	99,000
With Steering	10.8 Peak	6.0	4.7	.4	1.7	.9	3.7	2 to 30	14,900	61,000

Total Energy to be Stored:

- Nose Gear Unit: 96,000 Ft-lbs.
- Main Gear Unit:
  - For Gear Ext: 211,600 Ft-lbs.
  - For Bogie Fold: 121,000 Ft-lbs.

NOSE GEAR PUMP SIZE

Maximum Flow Requirements from Table IV

Gear Extension:	5.1
Steering:	7.0

Nominal Pump Size:

$$\frac{7}{80\% \times 86\%} = 10.4 \text{ GPM}$$

Eng RPM Pump eff.

Pump: PV012 produces 10.4 GPM at 13000 RPM and weighs 9 lbs.

MAIN GEAR PUMP SIZE

Maximum Flow Requirements from Table IV

Gear Extension:	8.9 GPM
Gear Retraction:	9.2 "
Bogie Fold:	18.5 "
Bogie Unfold:	15.0 "

Nominal Pump Size:

Gear Extension: $$\frac{8.9}{80\% \times 86\%} = 13.0 \text{ GPM}$$

Gear Retraction: $$\frac{9.2}{86\%} = 10.7 \text{ GPM}$$

Bogie Fold: $$\frac{18.5}{86\%} = 21.5 \text{ GPM}$$

Bogie Unfold: $$\frac{15.0}{80\% \times 86\%} = 21.8 \text{ GPM}$$

Because each system must power all functions for emergency extension the largest pump is required in each system.

PUMP: PV039 produces 21.8 GPM at 8500 RPM and weighs 22 lbs.

FLYWHEEL CHARGING

Nose Gear Unit:

Engine-driven pump flow available (from Table II, is 2.3 GPM.

$$\text{Motor displacement} = \frac{2.3 \times 231 \times 86\%}{13000 \text{ RPM}} = .0352 \text{ Cu. IN/REV}$$

$$\text{Yoke angle} = \sin^{-1} \frac{.0352}{.188 \times 2} = .0935 = 5^\circ \ 22'$$

Power = 4.2 H.P. at 3700 psi
Torque = 20.5 IN-LB. at 3700 psi

MAIN GEAR UNIT

Engine-driven pump flow available (from Table II, is 3.6 GPM.

$$\text{Motor displacement} = \frac{3.6 \times 231 \times 86\%}{8500} = .084 \text{ Cu. IN/SEC}$$

$$\text{Yoke angle} = \sin^{-1} \frac{.084}{.600 \times 2} = .0700 = 4^\circ$$

Power = 6.8 H.P. at 3700 psi
Torque = 49.5 IN-LBS at 3700 psi

FLYWHEEL DIMENSIONS

Sized for 10% Speed Reduction

ITEM	NOSE GEAR	MAIN GEAR	BOGIE FOLD
Energy Req., Ft-Lb	96,000	211,600	121,000
H.P. - Secs.	175	384	220
Dia., In	12	12	12
Vel., R/Sec	5,450	5,450	5,450
RPM	52,200	52,200	52,200
Tip Thickness, In	.08	.19	.11
Windage Loss, H.P.	1.65	1.65	1.65
Weight, LBS.	14	33	19
Shroud, Support Bearings, LBS.	9	10	9

WEIGHT COMPARISON

Stored Energy System - Configuration #2

Flywheel, LB.		14		33		19
Shroud, Supt., Brgs.		9		10		9
Pump, Flywheel		9		22		22
Gear Box, Ratio/	4:1		6:0		6:0	
Gear Box, Weight		7		12		10
Filter	10 GPM	6	20 GPM	10	20 GPM	10
		45		77		70

Pump, Engine, Lb.	14
Filter, Eng. - Pump, 11.9 GPM	6
Trunk Lines, Mn. Gr., 11.9 GPM = 116' x 5/8 =	30.5
N. Gr., 2.3 GPM = 112' x 5/16 =	10.5
	61

A/V Weight, 2 Systems: 2 x 45 = 90
2 x 77 = 154
2 x 70 = 140
2 x 61 = 122

TOTAL -- 506 LB.

CONFIGURATION #3

Description

This arrangement calls for a flywheel, a clutch actuated either by electric power or hydraulic power; a high ratio gear box, a reversing gear, a motor and planetary gear to spin up the flywheel, and suitable over-center linkage at each point of operation. If over-center linkage,which allows clutch engagement under the minor load of gear box and bellcrank inertia, is not used, clutch size and energy losses become quite large.

Figure 8 is a simplified schematic showing this concept applied to the B-70 aircraft. Figure 9 illustrates how this concept would be applied for main landing gear retraction. In theory, the over-center linkage could be the drag brace and also serve as the uplock. However, the travel of an acceptable drag brace is so great that it seems more efficient to retain the drag brace and the actuator to break it over-center. The nose gear is treated the same way. Wheel doors are too large to be held closed by torque at the hinge line, therefore, the door lock actuators and locks are retained.

The two hydraulic systems operating together share the load for normal operations of all the stored energy functions. Either system can power all functions but at reduced speed as the flywheels slow down below the 10% level.

A pressure controlled valve built into each charging motor cuts it off the line when pressure drops below 3500 psi. This allows steering, brakes, or bogie fold operations to take all flow from the engine-driven pump, restricting flywheel charging to times when other fluid requirements are less than engine pump output.

Because this approach requires structural and space modifications of some magnitude it is an illustrative configuration only and would be feasible only if designed into a new aircraft.

The addition of the stored energy concept using direct mechanical drive to the bogie fold operation conflicts with basic design requirements. It appears more practical for this study to redesign the bogie fold actuator to require no more flow than that which must be provided for brake operation. To keep the comparison between Configurations 1 and 3 as accurate as possible, the basic configuration is changed to incorporate the revised bogie actuator and is identified as configuration 1-A.

TRUNK LINES - CONFIGURATION 1-A

Flow Req. = 34.6 retraction

= 30.5 extension

Main Gr Lines = 35 GPM = 7/8 inch lines

= 116 ft x 7/8 O.D. = 57.4 lbs

Nose Gr Lines = 7 GPM = 3/8 inch tube

= 112 ft x 3/8 in. O.D. = 18 lbs

ENGINE PUMP - CONFIGURATION 1-A

The required pump size for the retraction and extension functions is:

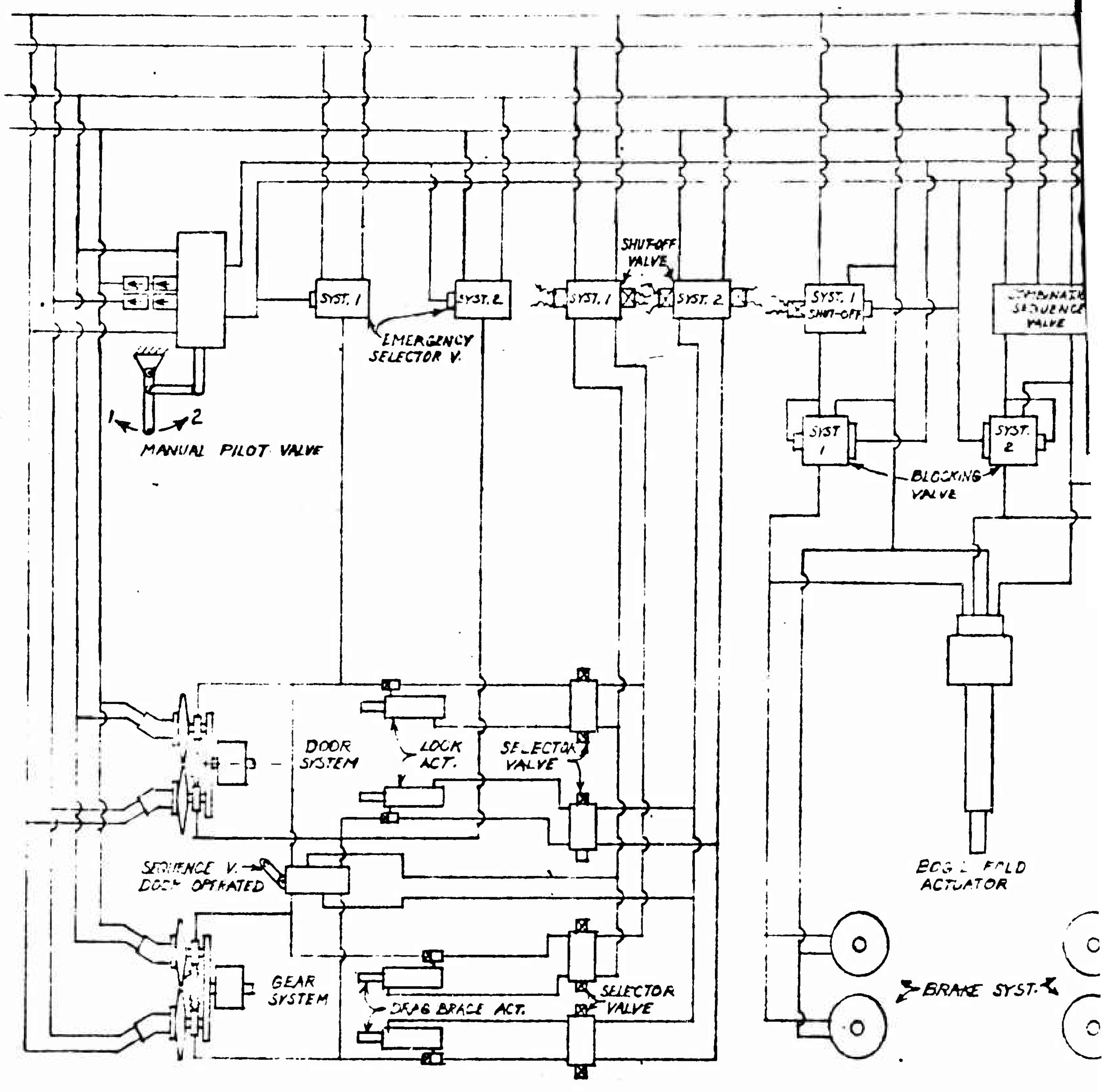
Retraction $= \frac{34.6}{86\%} = 40.2$ GPM

Extension $= \frac{30.5}{80\% \times 86\%} = 44.4$ GPM

Since the bogie folding requirements are reduced to equal those of the brakes, the required pump size is 44.4 GPM (See Table II).

Pump: PV104 at 6500 RPM = 44.4 GPM

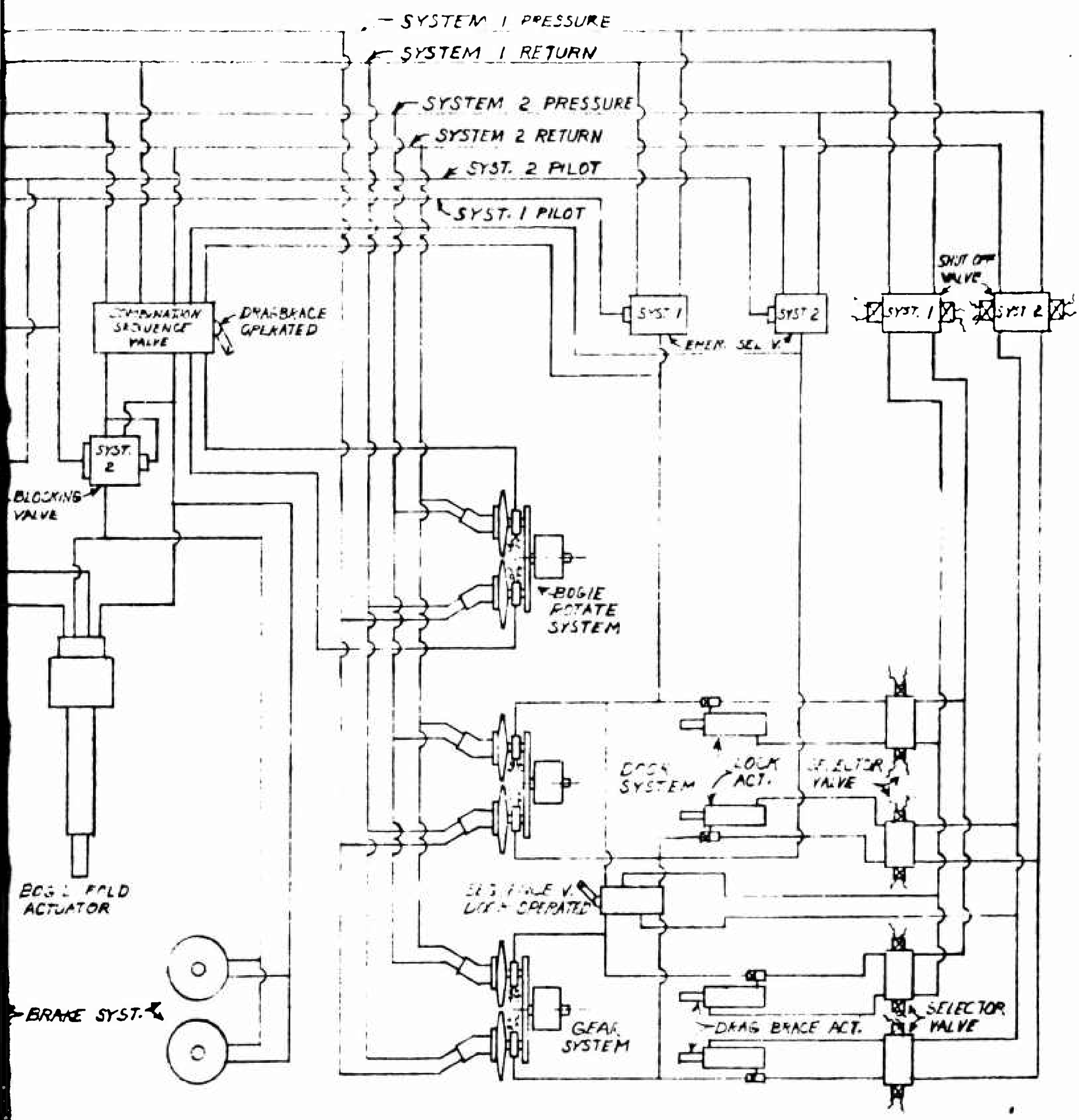
Weight = 38 lbs.



NOSE GEAR SYSTEM

I BOGIE SYSTEM

FIGURE 8
CONFIGURATION 3
SCHEMATIC DIAGRAM



BOGIE SYSTEM

MAIN GEAR SYSTEM

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CHECKED BY:	XB-70 MAIN GEAR MECH.	REPORT NO. NA-65-825-1
DATE: 28 JAN 66	ENERGY STORAGE SUBSTATION	MODEL NO. Figure 9

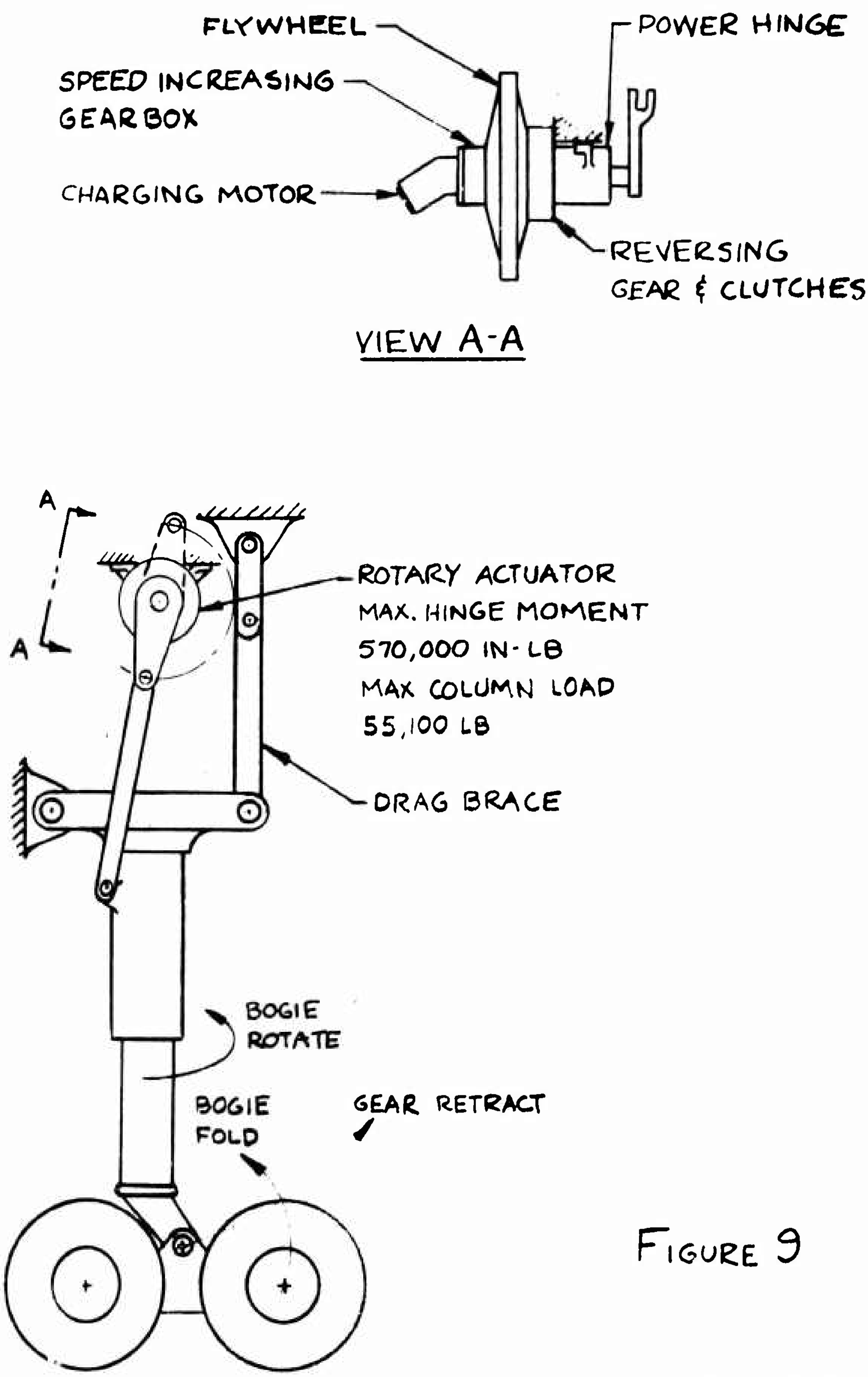


FIGURE 9

NOSE GEAR EXTENSION, CONFIGURATION #3

REQUIREMENTS AND ASSUMPTIONS

Engine RPM = 80% for gear extension
Gear extension time = 15.9 sec.
Time between consecutive operations = 30 secs.

Efficiencies:	Power Hinge	=	65%
	Gear Set	=	95%
	Overall Mechanical	=	95 x 95 x 65 = 58.7%
	Motor Volumetric	=	86% at 400F

Energy Requirement (fron load-stroke curve) = 601,000 in-lb.
Flywheel speed reduction = 10%

Gross Energy required to extend gear

$$\frac{601{,}000}{2 \times 6600 \times 58.7\%} = 77.6 \text{ HP-SEC/system}$$

Charging Rate

$$\frac{77.6}{30} = 2.6 \text{ HP}$$

Stored energy required

$$77.6 - (2.6 \times 15.9) = 36.2 \text{ HP-SECS}$$

Flywheel size

Tip thickness	=	.04
Dia.	=	8
Vel.	=	8,250 Rad/sec
	=	78,700 RPM
Weight	=	3 lb
Weight, Shroud	=	4.2 lb
Windage	=	.8 HP

Motor Size

$$\text{Nominal} = \frac{2.6 + .8}{80\% \times 86\%} = 4.95 \text{ HP}$$

PF003 at 3500 psi = 4.95 HP at 11,700 RPM = 2.4 GPM

Weight = 3.6 lb

Gear Ratios

Flywheel to Motor (Gear Set #1) $= \frac{78,700}{11,700} = 6.73 : 1$

Motor to Power Hinge (Gear Set #2) $= \frac{11,700}{4,500} = 2.6 : 1$

Power Hinge $= \frac{4500 \times 15.9}{60} : 1/2 = 2380 : 1$

NOSE GEAR DOOR CLOSING

REQUIREMENTS AND ASSUMPTIONS

Engine RPM = 80% for gear extension
Door closing time = 3 secs.
Time between consecutive operations = 10 sec.
Efficiencies: Power Hinge = 65%
Gear Sets = 95%
Overall Mechanical = 65 x 95 x 95 = 58.7%
Motor Volumetric = 86% at 400F
Energy requirement (load-stroke curve) = 75,600 in-lb
Flywheel speed reduction = 10%

Gross Energy to close door

$$\frac{75,600}{2 \times 6600 \times 58.7\%} = 9.8 \text{ HP-SECS/system}$$

Charging Rate

$$\frac{9.8}{10} = 1 \text{ HP}$$

Stored energy required

9.8 - (1 x 3 sec.) = 6.8 HP-Secs

Flywheel Size

Tip thickness	=	.010
Dia.	=	6.4 in.
Vel.	=	10,500 rad/sec
	=	10,000 RPM
Weight	=	.52 lb
Weight, Shroud, etc.	=	2.6 lb
[illegible]	=	.6 HP

Motor Size

Nominal: $\dfrac{1 + .6}{80\% \times 86\%} = 2.4$ HP

PF003 at 3500 PSI = 2.4 HP

at 5700 RPM = 1.2 GPM

Gear Ratios

Flywheel to Motor (Gear Set #1) $= \dfrac{10,500}{5,700} = 1.84 : 1$

Motor to Power Hinge (Gear Set #2) $= \dfrac{5,700}{4,000} = 1.42 : 1$

Power Hinge $= \dfrac{4000 \times 3}{60} : 1/2 = 400 : 1$

MAI[illegible] EXTENSION, CONFIGURATION #3

REQUIREMENTS AND ASSUMPTIONS

Engine RPM Gear retraction = 100%; extension = 80%
Gear extension time = 12.5 sec.
Time between consecutive operations = 30 sec.

Efficiencies:	Power Hinge	=	65%
	Gear Train	=	95%
	Overall mechanical	=	65 x 95 x 95 = 58.7%
	Motor overall	=	86% at 400°F

Energy (from load-stroke curves) = 735,460 in-lb
Energy storage flywheels - 10% speed reduction normal.

Gross Energy to Extend Gear

$$\frac{735,460}{2 \times 6600 \times 58.7\%} = 96 \text{ HP-SEC/system}$$

Charging Rate

$$\frac{96}{30} = 3.2 \text{ HP}$$

Stored Energy Required

$$96 - (3.2 \times 12.5) = 56 \text{ HP-SEC}$$

Flywheel Size

Tip thickness	=	.05 in.
Dia.	=	8.4 in.
Vel.	=	8,000 rad/sec
	=	76,400 RPM
Weight	=	4.5 lb
Weight, Shroud, etc.	=	4.3 lb
Windage	=	.85 HP

Motor Size

$$\text{Nominal size} = \frac{3.2 + .85}{80\% \times 86\%} = 5.9 \text{ HP}$$

PF003 at 14,000 RPM = 2.9 GPM

Weight - 3.6 lb

Gear Ratios

$$\text{Flywheel to Motor (Gear Train \#1)} = \frac{76,400}{14,000} = 5.45 : 1$$

$$\text{Motor to Power Hinge (Gear Train \#2)} = \frac{14,000}{5,000} = 2.8 : 1$$

$$\text{Power Hinge} = \frac{5000 \times 12.5}{60} : \frac{1}{2} = 2085 : 1$$

MAIN GEAR DOOR CLOSING

REQUIREMENTS AND ASSUMPTIONS

Engine RPM = 80%
Door Close Time - 3 sec.
Time between consecutive operations - 10 sec.

Efficiencies:	Power Hinge	=	65 %
	Gear Set	=	95%
	Overall Mechanical	=	65 x 95 x 95 - 58.7%
	Motor Volumetric	=	86% at temperature

Energy (from load-stroke curves) = 112,000 in-lb.
Flywheel speed reduction = 10%

Gross Energy to Extend Gear

$$\frac{112,000}{2 \times 6600 \times 58.7\%} = 14.4 \text{ HP-SEC/system}$$

Charging Rate

$$\frac{14.4}{10} = 1.44 \text{ HP}$$

Stored Energy Required

$$14.4 - (1.4 \times 3) = 10.2 \text{ HP-SEC}$$

Flywheel Size

Tip thickness	= .010
Dia.	= 8
Vel.	= 8,300 rad/sec
	= 79,000 RPM
Weight	= .8 lb
Weight, Shroud, etc.	= 4.0 lb
Windage	= .8 HP

Motor Size

$$\text{Nominal Size} = \frac{1.44 + .8}{80\% \times 86\%} = 3.26 \text{ HP}$$

PF003 at 3500 psi = 3.26 HP
= 1.6 GPM at 7700 RPM

Weight = 3.6 lb

Gear Ratios

$$\text{Flywheel to Motor (Gear Set \#1)} = \frac{79{,}000}{7{,}700} = 10.2 : 1$$

$$\text{Motor to Power Hinge (Gear Set \#2)} = \frac{7{,}700}{4{,}500} = 1.71 : 1$$

$$\text{Power Hinge} = \frac{4500 \times 3}{60} : \frac{1}{2} = 450 : 1$$

BOGIE ROTATE, CONFIGURATION #3

REQUIREMENTS AND ASSUMPTIONS

Engine 80% at gear extension
Bogie rotate time - 5.5 secs.
Time between consecutive operations = 30 secs.
Efficiencies: Power Hinge = 65%
Gear Set = 95%
Overall Mechanical = 65 x 95 x 95 = 58.7%
Motor Volumetric = 86% at temperature
Energy requirement (Table) = 5.5 GPM at 3500 psi for 5.5 secs.

$$= \frac{5.5 \times 3500 \times 5.5}{2 \times 17.4 \times 58.7\%} = 53 \text{ HP-SECS/syst.}$$

Charging Rate

$$\frac{53}{30} = 1.8 \text{ HP}$$

Stored Energy Required

$$53 - (1.8 \times 5.5) = 43 \text{ HP-SECS.}$$

Flywheel Size

Tip Thickness	=	.040
Dia.	=	8
Vel.	=	8,250 rad/sec.
	=	78,700 RPM
Weight	=	3.6
Weight, shroud, etc.	=	4.2
Windage	=	.8 HP

Motor Size

$$\text{Nominal} = \frac{1.8 + .8}{80\% \times 86\%} = 3.8 \text{ HP}$$

PF003 at 9000 RPM = 1.85 GPM

= 3.8 HP at 3500 psi

Weight = 3.6 lb

Gear Ratios

Flywheel to Motor (Gear Set #1) $\frac{78,700}{9,000} = 8.75 : 1$

Motor to Power Hinge (Gear Set #2) $\frac{9,000}{4,500} = 2 : 1$

$$\text{Power Hinge} = \frac{4500 \times 5.5}{60} : \frac{1}{2} = 825 : 1$$

ENGINE PUMP - CONFIGURATION #3

TABLE V

ITEM	FLYWHEEL CHARGING GPM	OTHER REQUIREMENTS GPM
Nose Gear Ext.	2.4	
Nose Gear Door	1.2	
Nose Gear Steering		7.0
Main Gear Ext.	(2.9) 5.8	
Main Gear Door	(1.6) 3.2	
Bogie Rotate	(1.9) 3.8	
Bogie Fold		(10.0) 20.0
Brakes		(10.8) 21.6
Simultaneous Flow Required	16.4	28.6

Nominal Pump Size $= \frac{28.6}{80\% \times 86\%} = 41.6$ GPM

A PV104 pump at 6100 RPM = 41.6 GPM

Weight - 38 lbs

TABLE VI

POWER HINGE SIZING

ITEM	NOSE GEAR	NOSE GEAR DOOR	MAIN GEAR	MAIN GEAR DOOR	BOGIE ROTATE
Max. Torque (load-stroke-cruve)	286,000	28,000	570,000	52,000	150,000
Des. Life, Cycles	2,900	5,800	2,900	5,800	2,900
Angular Travel per Cycle	360	360	360	360	80
Max. Dia.	8	4	10	5	6
No. Stress Cycles	140	79	165	94	110
Tooth Stress/cycle	406,000	459,000	479,000	545,000	71,000
X^{1}_{4} (over-center cor. factor)	101,000	115,000	120,000	136,000	71,000
% of Ult. Torque	22	22	22	22	23
Ult. Torque	1,300,000	127,000	2,590,000	236,000	652,000
Ult. Torque per Inch	190,000	41,500	300,000	70,000	97,000
Length, in.	6.8	3.1	8.6	3.4	6.7
Weight/in.	8.4	2.2	12.0	3.3	4.7
Weight, LB.	57	7	103	11	32

TABLE VII

CLUTCH AND GEAR SET SIZE AND WEIGHTS

Location	Peak Torque (Load-Stroke) In-Lbs	Power Hinge Ratio	Clutch Torque Ft-Lb.	Gear #2 $\frac{T \times RPM}{63,025}$ H.P.	Gear # 2 Ratio	Torque at O'Run Clutch Ft-Lbs	Weight		
							Clutch	O'Run Clutch Lb.	Gear Set
Nose Gear	286,000	2380 : 1	10.0	22.2	2.6 : 1	3.8	4	0.2	5
Nose Gear Door	28,000	400 : 1	5.8	6.3	1.4 : 1	4.2	[illegible]	0.2	2
Main Gear	570,000	2085 : 1	22.8	60.9	2.3 : 1	8.2	6	0.4	13
Main Gear Door	52,000	450 : 1	9.6	14.1	1.7 : 1	5.7	4	0.3	3
Bogie Rotate	150,000	825 : 1	15.2	26.1	2 : 1	7.6	5	0.3	6

TABLE VIII

WEIGHT COMPARISON

All items which are different in the two configurations.

ITEM	CONFIGURATION 1-A NO. REQ.	CONFIGURATION 1-A UNIT WT. LBS.	CONFIGURATION 1-A TOTAL WT. LBS.	CONFIGURATION 3 NO. REQ.	CONFIGURATION 3 UNIT WT. LBS.	CONFIGURATION 3 TOTAL WT. LBS.
Emer. Selector Valve	1	3.0	3.0	6	2.5	15.0
10 Port Valve	2	86.0	172.0	0	-	-
Emer. Selector Valve	1	10.0	10.0	0	-	-
N. Gear Sel. Valve	1	3.0	3.0	3	3.0	9.0
Door Sel. Valve	3	5.2	15.6	8	3.0	24.0
M. Gear Sel. Valve	2	5.4	10.8	5	3.0	15.0
Bogie Rotate Valve	2	5.4	10.8	0	-	-
Gear Sequence Valve	3	4.9	14.7	3	5.3	15.9
N. Gear Act.	1	47.5	47.5	0	5.3	-
N. Gear Uplock Act.	1	3.3	3.3	0	-	-
N. Gear Drag Brace Act.	1	17.0	17.0	2	9.0	18.0
N. Gear Door Act.	1	26.9	26.9	0	-	-
N. Gear Door Lock Act.	1	3.6	3.6	2	3.6	7.2
Bogie Rotate Act.	4	42.0	168.0	0	-	-
Bogie Rot. Pin Pull Act.	4	41.0	164.0	0	-	-
Bogie Rot. Latch Act.	2	2.0	4.0	0	-	-
Main Gear Door Act.	4	27.3	109.2	0	-	-
Main Gear Door Lock Act.	2	3.8	7.6	4	3.8	7.6
Main Gear Act.	2	207.4	414.8	0	-	-
Main Gear Uplock Act.	2	4.7	9.4	0	-	-
Main Gear Drag Brace Act.	2	9.0	18.0	4	9.0	36.0
Bogie Sequence Valve Lines	2	18.0	36.0	2	19.5	39.0
Trunk-Eng. - Main Gear	2	57.4	114.8	2	42.9	85.8
Subsystem	1	81.1	81.1	1	31.1	31.1
Engine-Driven Pump	1	38.0	38.0	1	38.0	38.0
Flywheels	0	-	-	10	Σ	24.8
Shrouds, Brgs, etc.	0	-	-	10	Σ	38.6
Power Hinge	0	-	-	5	Σ	210.0
Motors	0	-	-	10	3.6	36.0
Linkages	0	-	-	5	Σ	176.0
Powered Clutches	0	-	-	8	Σ	72.0
Clutch Actuators	0	-	-	16	2.0	32.0
O'Run Clutch	0	-	-	16	Σ	4.8
Gear Sets	0	-	-	40	Σ	255.0
TOTALS			1503.1			1190.8
DIFFERENCE						312.3

SYSTEM PERFORMANCE ANALYSIS (ANALOGUE STUDIES)

The system performance analyses of the energy storage flight control system which utilizes hydraulic techniques have been completed and the reduction of the analog computer generated data has been initiated. In addition, the system configuration utilizing a toroidal mechanical servo, rotary transmission shaft and power hinges and a flywheel-induction motor has also been evaluated on the analog computer.

Although the data reduction on the hydraulic system has not been completed, preliminary information indicates the general capabilities of the system. The following paragraphs and graphs will illustrate the expected performance levels of such hydraulic applications. (figures 10 through 18.)

This performance analysis utilized the F-100 horizontal stabilizer actuation system characteristics and requirements for the test configuration. The linear hydraulic actuator is a dual tandem actuator utilizing 3,000 psi hydraulics. For this investigation, the system was assumed to be one hydraulic source driving a single actuator with the hinge moment reduced by 50 percent.

As described in the first quarterly report, the hydraulic system application consisted of an engine-driven pump, a motor-pump shafted to a flywheel, a speed control loop and a linear hydraulic actuator. A schematic of such a system is shown in figure 10. The contribution of the energy storage portion of this system is in satisfying the high flow demands of the actuator.

Since the flywheel is assumed to be attached directly to the motor-pump and hence rotating at the same angular velocity, the flywheel inertias utilized in this study can be specified in terms of horsepower-seconds.

$$\text{HP-SECS} = \frac{I W_F^2}{550 \times 12}$$

where I is in IN LB SEC^2 and W_F is in RAD/SEC. For example, for I = 20 and W_F = 100,

Flywheel Energy = 30.3 hp-secs.

The actual choice of a flywheel to provide such an energy source would be dependent upon its maximum useable material strength and be made through the tables of optimum flywheel sizes. Flywheel location and gearing would determine the required inertia and angular velocity.

The nominal size of the engine-driven pump was determined by the hydraulic actuator requirements. With the incorporation of the flywheel-motor-pump, it was then determined to what extent the engine-driven pump could be reduced in capacity while maintaining satisfactory system performance.

SYSTEM SCHEMATIC

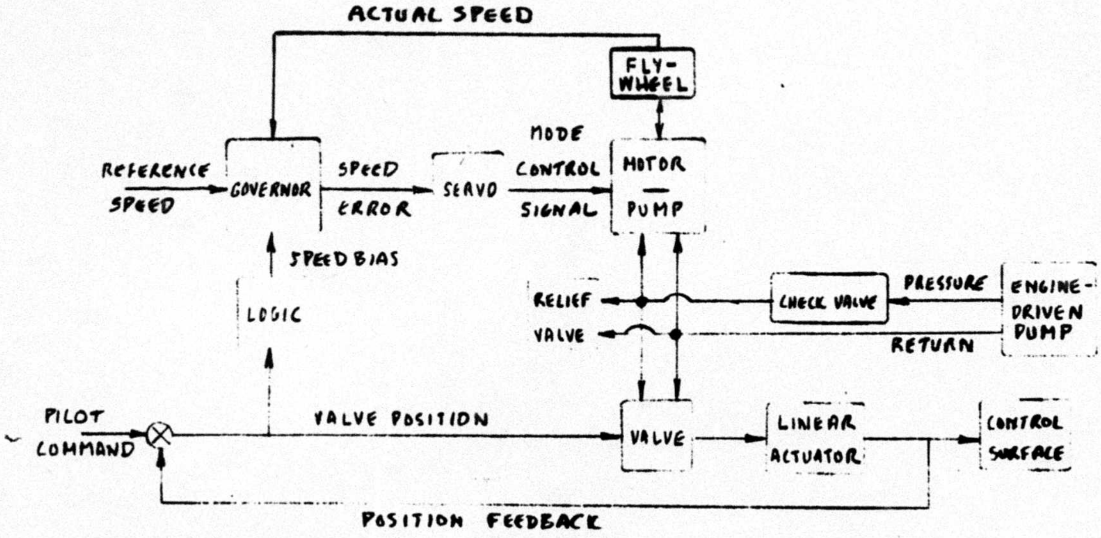
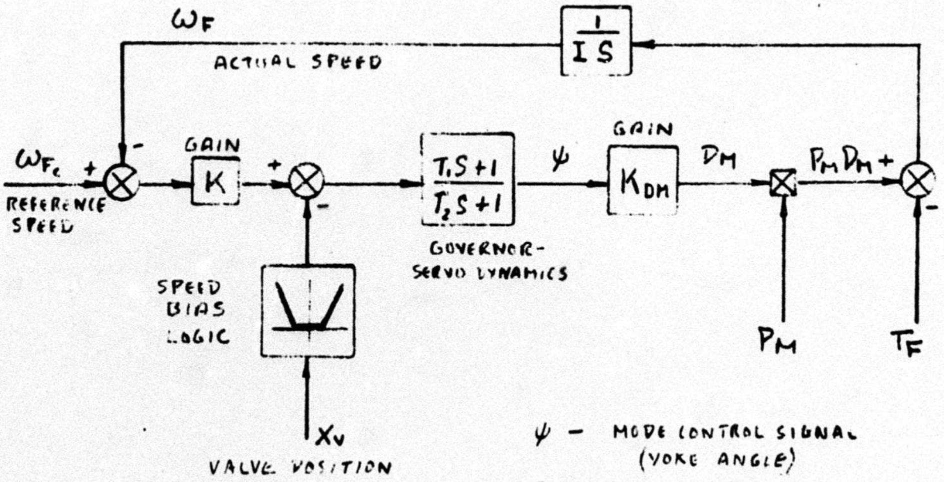


FIGURE 10

SYSTEM BLOCK DIAGRAM



ψ – MODE CONTROL SIGNAL (YOKE ANGLE)

D_M – MOTOR-PUMP DISPLACEMENT

P_M – MOTOR-PUMP PRESSURE

T_F – TORQUE LOSSES (WINDAGE, FRICTION)

I – FLYWHEEL INERTIA

FIGURE 11

An examination was first made of the flywheel-motor-pump-speed control loop to determine the effects of loop gain and torque losses. A block diagram of this loop is shown in figure 11. The speed control transfer function, using the terminology in figure 11, is found to be:

$$\frac{\omega_F}{\omega_{F_c}} = \frac{P_M K_{D_M} K T_1 \left(s + \frac{1}{T_1}\right)}{IT_2 \left[s^2 + \left(\frac{1}{T_2} + \frac{P_M K_{D_M} K T_1}{IT_2}\right) s + \frac{P_M K_{D_M} K}{IT_2}\right]}$$

First of all, it is to be noted that increasing the gains K_{D_M} and K has the same effect as decreasing I (flywheel inertia). Using the following system gains,

$$K = .004 \text{ rad/rad/sec}$$

$$K_{D_M} = 2 \text{ in}^3/\text{rad}^2$$

$$T_1 = .4 \text{ sec}$$

$$T_2 = .2 \text{ sec}$$

$$P_M = 3{,}000 \text{ PSI}$$

The transfer function denominator becomes:

$$s^2 + \left(5 + \frac{48}{I}\right) s + \frac{120}{I}$$

For I = 20 in-lb-sec^2 the roots are about:

$$(s + 6.5)\ (s + .9)$$

While for I = 2,000 they are:

$$(s + 5)\ (s + .012)$$

Hence, as might be expected increasing the inertia (or decreasing K_{D_M} K) slows the loop response. Increasing loop response however means a higher rate of speed loss or larger speed fluctuations under transient conditions.

Assuming constant torque losses, e.g., due to windage or friction, a determination of motor-pump flow demands for quiescent conditions can be made. Again using figure 11 terminology the flow into the motor from the engine-driven pump to support the torque losses is:

$$Q = \frac{T_F}{P_M} \left[\omega_{F_c} - \frac{T_F}{P_M K_{D_M} K}\right]$$

Assuming ω_{F_C} = 100 rad/sec and P_M = 3,000 PSI, figure 12 shows the flow as a function of $K_{D_M}K$ and T_F. Likewise the steady state speed loss given by:

$$\Delta\omega_f = \frac{T_F}{P_M K_{D_M} K}$$

is illustrated in the same graph. It can be seen that for a given torque, T_F, increasing the gain $K_{D_M}K$ results in a raising of flow requirements during quiescent conditions while at the same time lowering the speed loss.

For speed losses due to transient conditions the flow demand seen by the engine-driven pump is:

$$Q = K_{D_M}K \; (\omega_{F_C} - \Delta\omega_f)\,\Delta\omega_f$$

A plot of Q versus $\Delta\omega_f$ and $K_{D_M}K$ is presented in figure 13. It can be seen that for any given speed loss a higher $K_{D_M}K$ gain results in a higher flow demand. Using these data, a plot of engine-driven pump excess flow capability as a function of flywheel speed loss is shown in figure 14. This flow capability must be biased, however, to take into account the steady state flow losses which total to about 9.5 in^3/sec (6.5 in^3/sec for the motor-pump and 3 in^3/sec for the engine-driven pump). This bias is shown by the dashed, straight line. For a given speed loss, the excess flow available for flywheel acceleration is represented by the difference between the constant flow loss line and the appropriate pump size curve. Hence, it is apparent, for instance, that the lower pump size (15 in^3/sec) cannot generally recover from speed losses in excess of 8 rad/sec. Recovery from momentary speed losses in excess of 8 rad/sec can be made if the existing pressure under such flow conditions is large enough to overcome the constant torque loss, T_F. This is determined by the relationship

$$P_M > \frac{T_F}{D_M}$$

The significance of these graphs is that because of these flywheel-motor-pump losses, which must be carried by the engine-driven pump in addition to its own and the actuator losses, very little size reduction is feasible in the engine-driven pump. Decreasing the gain $K_{D_M}K$, decreases the flow requirements but increases the steady state speed loss. For a 100 rad/sec reference speed, a maximum 25 rad/sec speed loss is allowed, even for transient conditions. The effect of increasing the reference speed is to increase the flow requirements.

The nominal pump capacity for this application is about 30 in^3/sec. Based upon the anticipated losses and transient conditions during operation, the pump size can be reduced only to an extent consistent with the allowable speed reduction.

FLOW DEMAND & SPEED LOSS VS. SPEED CONTROL GAIN

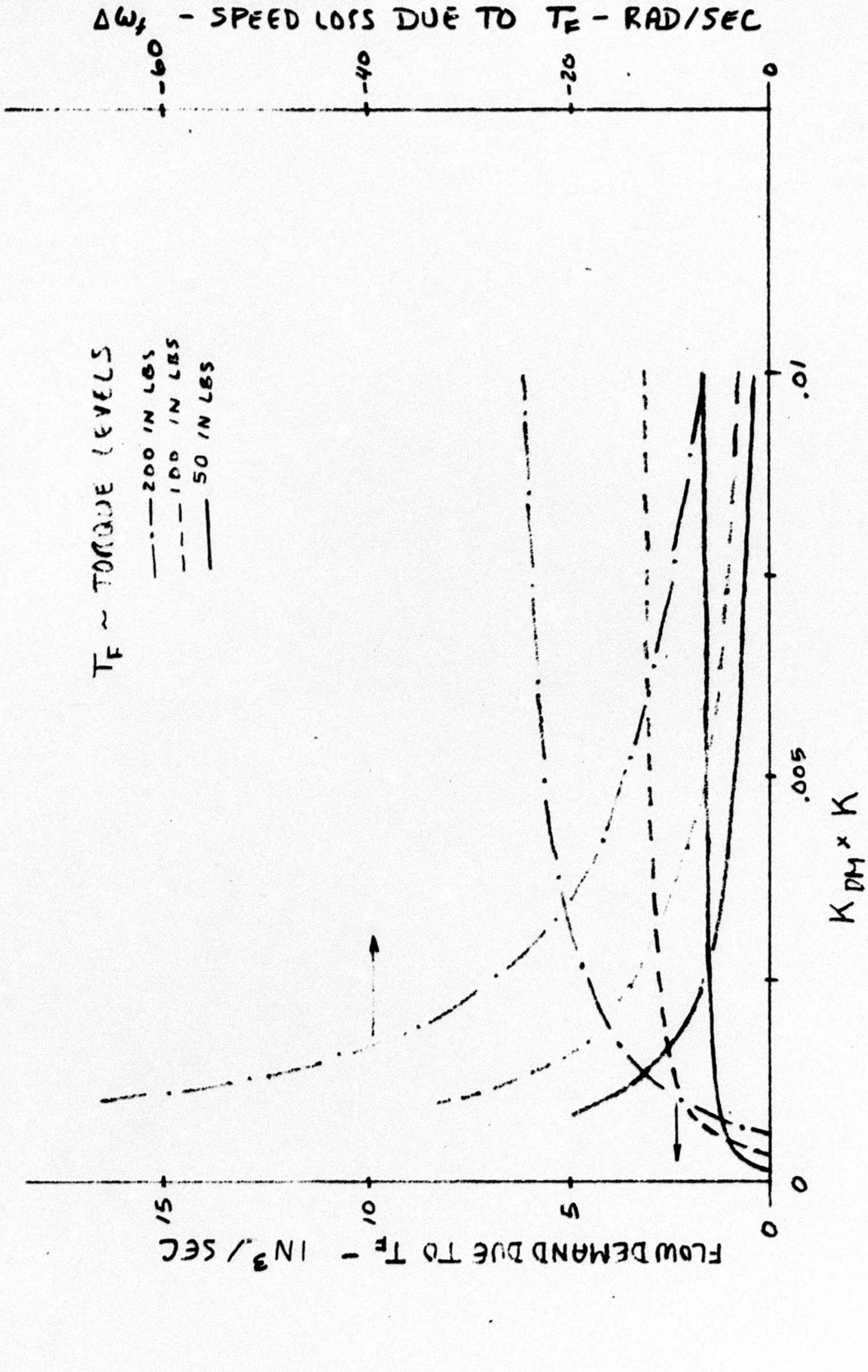


FIGURE 12

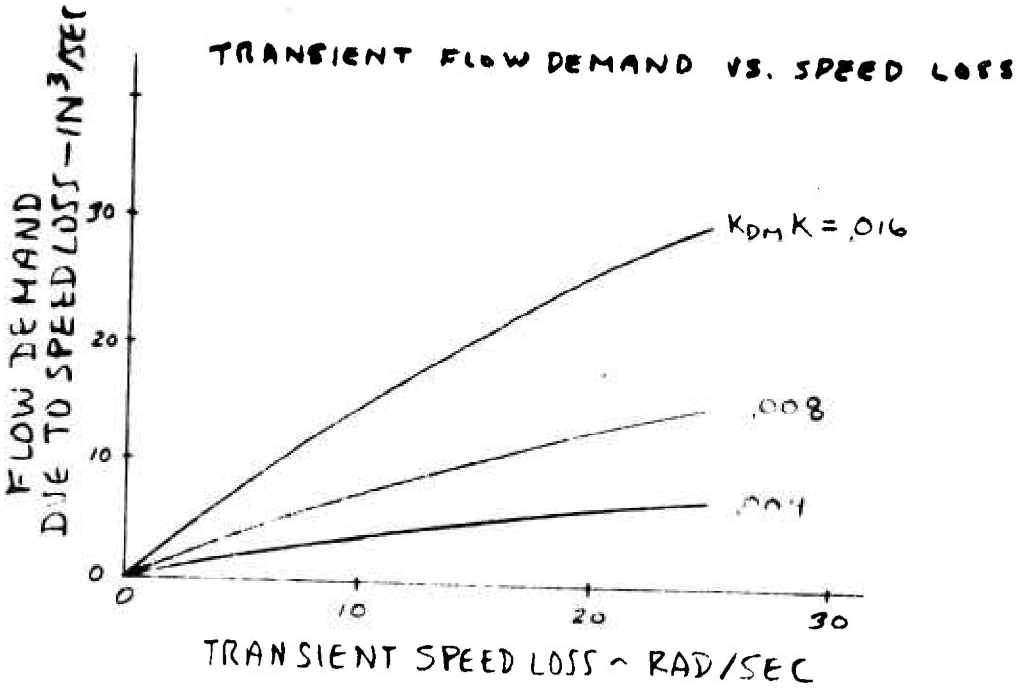


FIGURE 13

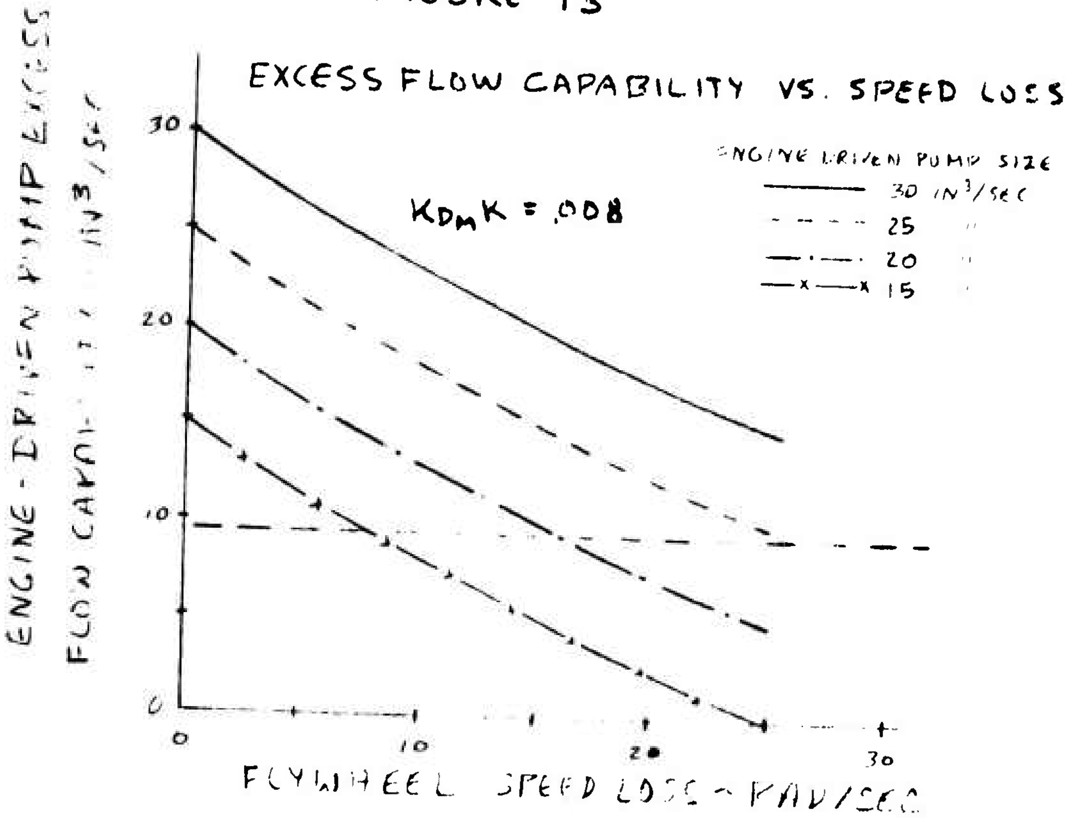


FIGURE 14

In the analog computer data runs various duty cycles were utilized to examine system performance. These were a terrain following duty cycle and sinusoidal duty cycles which ranged in frequency from 1/8 to 1 cps and in amplitude from ± 1/4 inch to ± 1.0 inches. At the flight condition utilized (.95 Mach at S.L.) stall hinge moment 500×10^3 in lb (at hinge line) is encountered at about 1.3 inches of actuator travel.

Data reduction has not yet been completed to the extent that dynamic performance comparisons (frequency response) between the nominal and the flywheel-motor-pump systems can be made. However, those data available indicate the relative sizing required for the engine-driven pump and the flywheel for allowable flywheel speed losses. For the terrain following duty cycle (figure 15) all of the pump sizes evaluated coupled with the various flywheel sizes resulted in 12 rad/sec or less speed losses. This is shown in figure 16. Also depicted on the graph are the results obtained when the duty cycle was modified to result in larger surface deflections (amplitude increased threefold, hinge moment reduced by 75%). Under this condition the 15 in^3/sec pump size coupled with the 20 in lb sec^2 becomes unacceptable.

The sinusoidal duty cycle data are presented in figures 17 and 18. The indicated frequencies and amplitudes are the duty cycle inputs. As might be expected, the more severe the duty cycle the more stringent is the pump size requirement for a given flywheel size. Figure 17 shows the system speed losses for a flywheel size of 200 in lb sec^2. The difference in these data results from those shown in figure 17 is the inclusion of curve segments characterized by stripes. These portions of the curve reflect system performance which is actually unacceptable if the duty cycle is allowed to exist indefinitely. However, such duty cycles can be expected to last only for relatively short times because of the resulting severe air vehicle responses. For this reason it was decided if the flywheel speed loss during this short time was less than three quarters of those losses indicated in figure 14 the system concerned was considered acceptable. It will be noted the 15 in^3/sec pump is still unacceptable for the 1 cps, ± 1 in amplitude duty cycle.

Such preliminary examination of the data indicates that commanded rates are more critical than hinge moment for the hydraulic system application. Continued effort is being made to further define the relationships between acceptable system size and duty cycle characteristics. In addition, further examination of the data will allow the description of the various hydraulic systems frequency response characteristics.

TERRAIN FOLLOWING DUTY CYCLE

ACTUATOR DISPLACEMENT (INCHES) VS. TIME

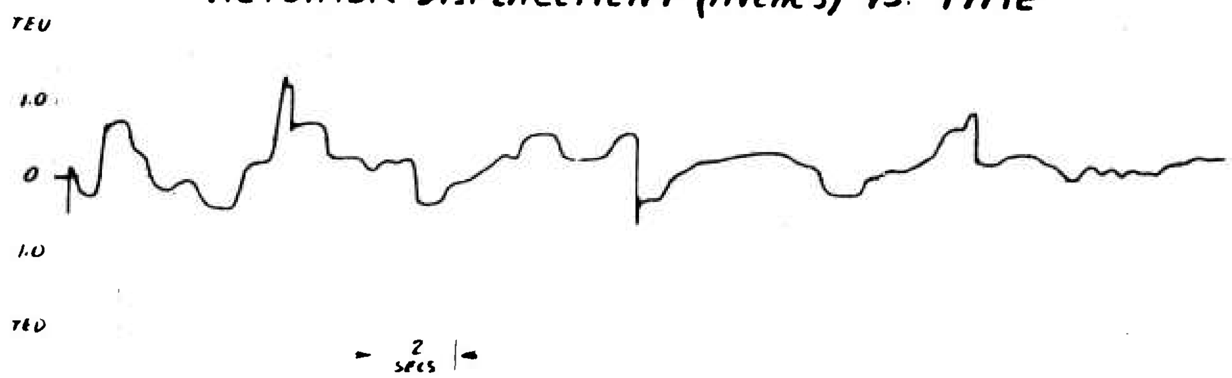


FIGURE 15

FLYWHEEL INERTIA VS FLYWHEEL SPEED LOSS FOR TERRAIN FOLLOWING

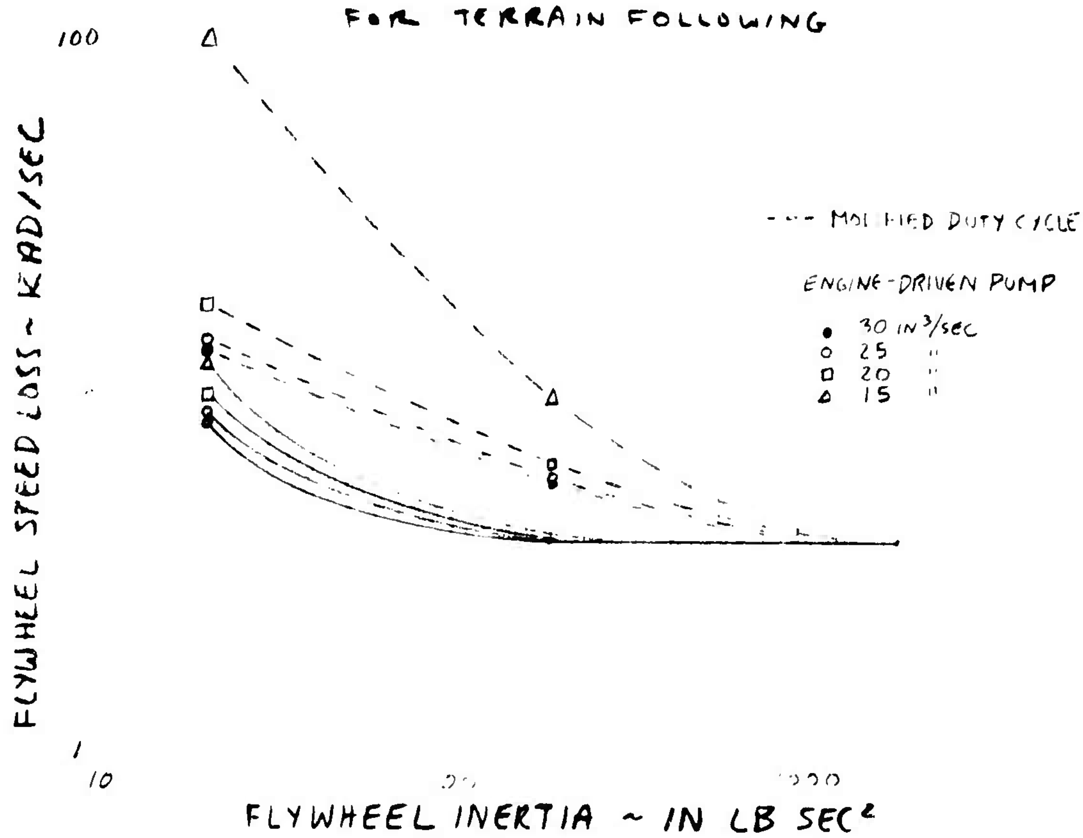


FIGURE 16

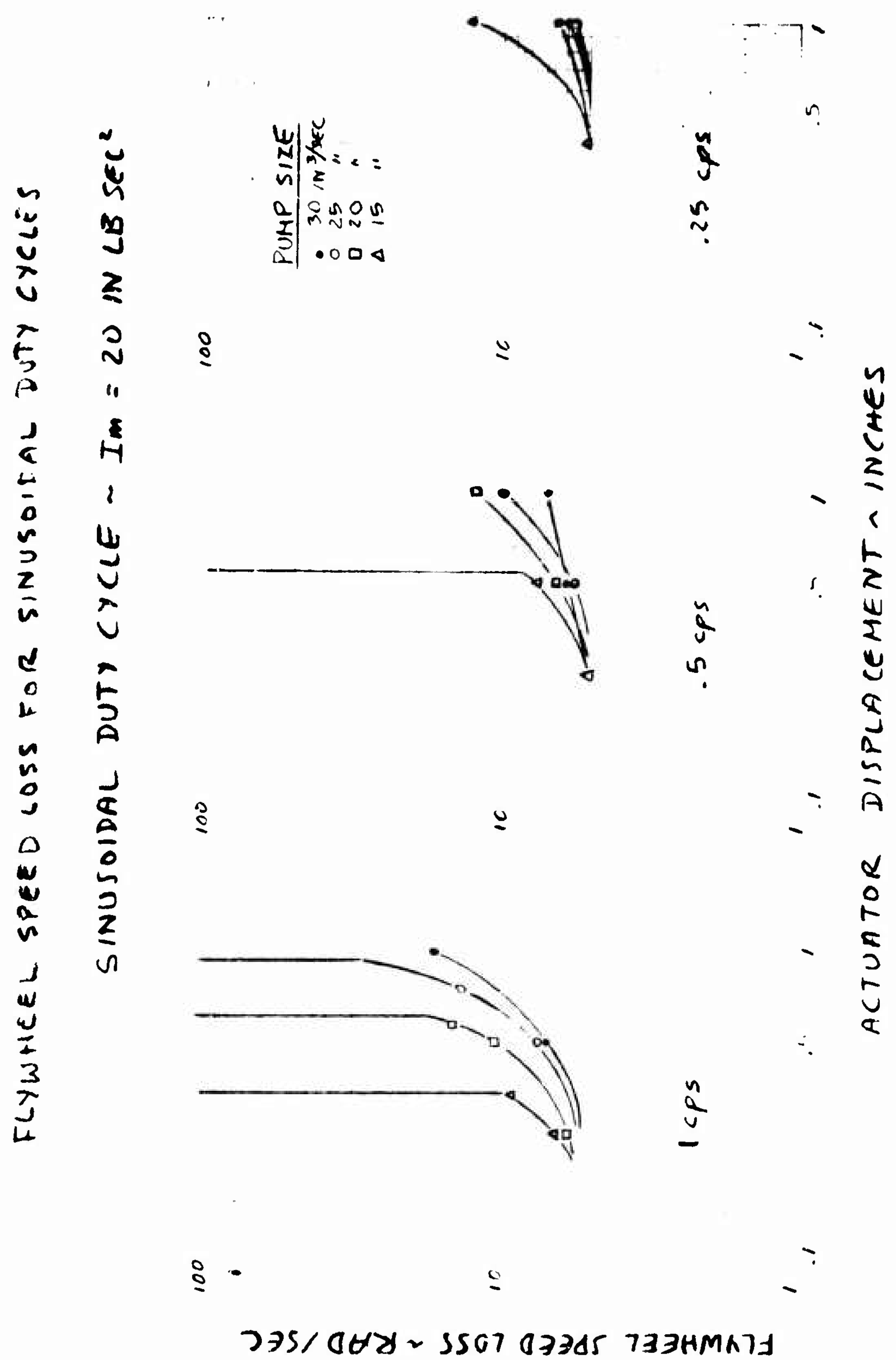

FLYWHEEL SPEED LOSS FOR SINUSOIDAL DUTY CYCLES
SINUSOIDAL DUTY CYCLE ~ I_m = 20 IN LB SEC²
PUMP SIZE
30 IN³/SEC
25 "
20 "
15 "
.25 cps
.5 cps
1 cps
100
10
.5
ACTUATOR DISPLACEMENT ~ INCHES
FLYWHEEL SPEED LOSS ~ RAD/SEC

FIGURE 17

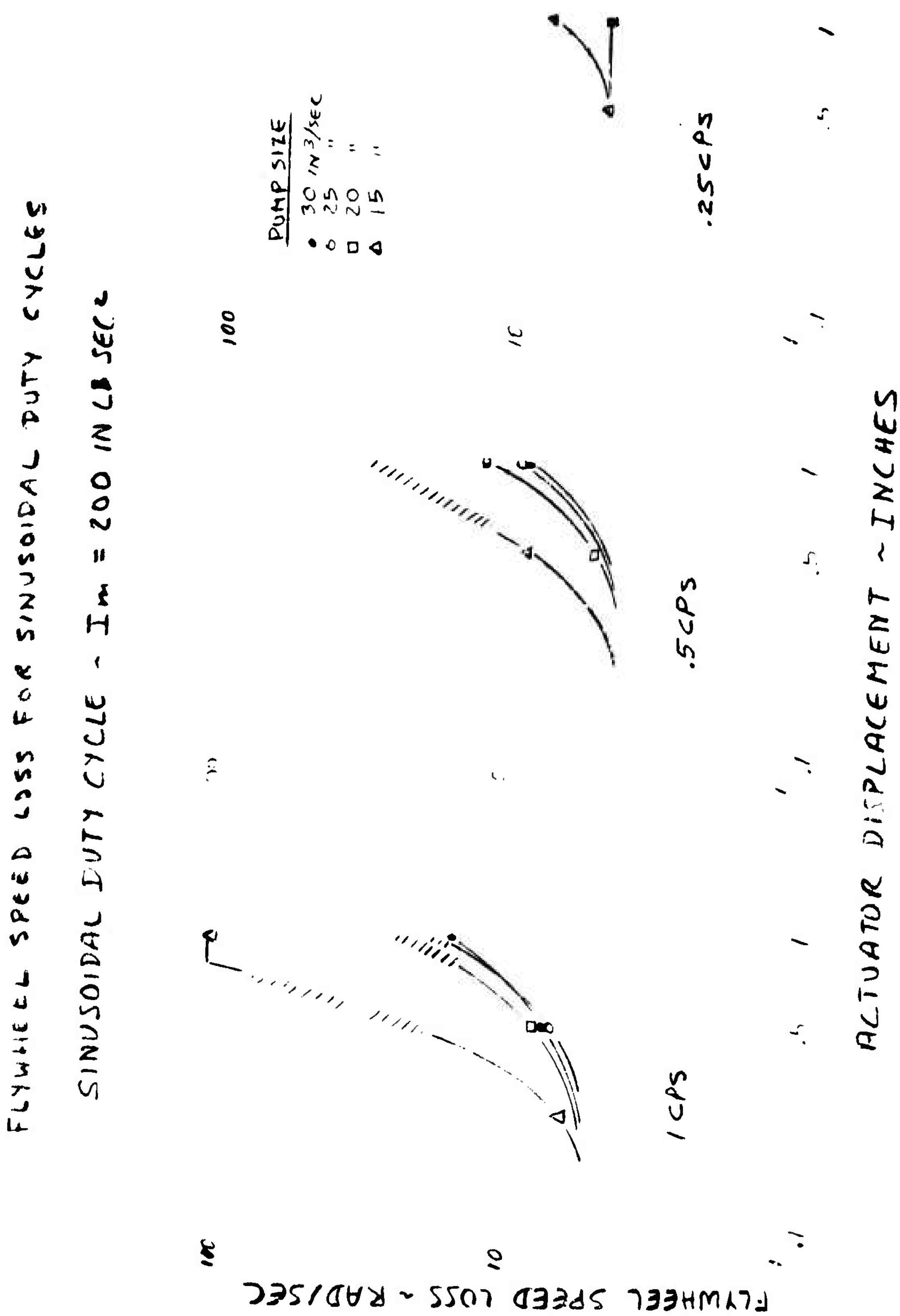

FLYWHEEL SPEED LOSS FOR SINUSOIDAL DUTY CYCLES
SINUSOIDAL DUTY CYCLE ~ I_m = 200 IN LB SEC²
PUMP SIZE
• 30 IN³/SEC
○ 25 "
□ 20 "
△ 15 "
.25 CPS
.5 CPS
1 CPS
100
10
ACTUATOR DISPLACEMENT ~ INCHES
FLYWHEEL SPEED LOSS ~ RAD/SEC

FIGURE 18